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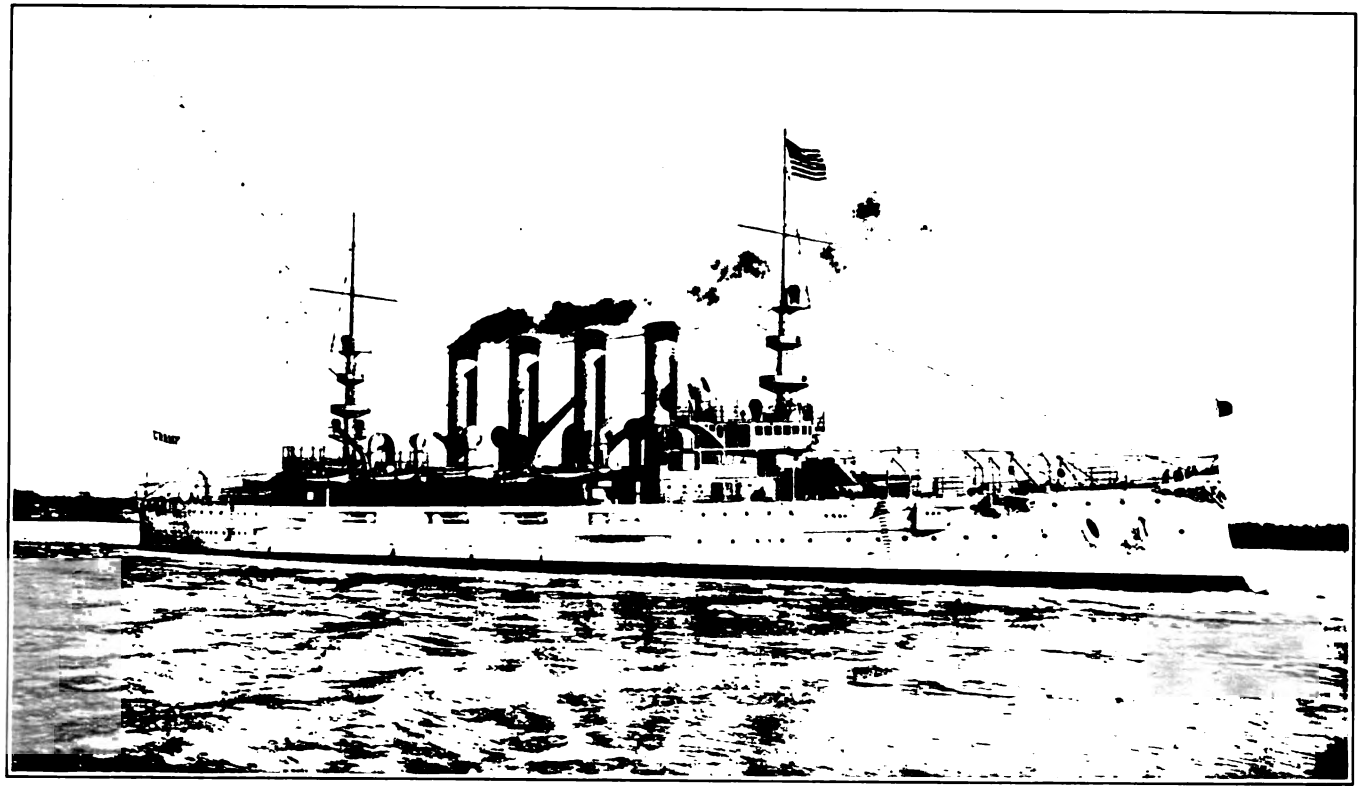

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THE U. S. S. *PENNSYLVANIA*.

BY LEWIS HOBART KENNEY, B. S., ASSOCIATE.

The U. S. armored cruiser *Pennsylvania* was built by the William Cramp and Sons Ship and Engine Building Company, Philadelphia, Pennsylvania. The contract was signed January 10, 1901, for the ship to be completed in thirty-six (36) months. This time was extended to August 10, 1904, and again to December 10, 1904. The hull was launched August 22, 1903. On September 15th and 16th, 1904, the port and starboard engines, respectively, were given a dock trial. The maximum pressure in the H.P. valve chest was 250 pounds and the corresponding revolutions per minute 88. There was no preliminary trial, and the ship left the yard of the builders November 12, 1904, for the official trial over the Cape Ann course. After leaving the Delaware Capes the ship proceeded to the Brooklyn Navy Yard, where she was dry-docked and painted, and also received her supply of coal. Here the lumps were picked out of the ship's supply of coal and bagged for the official trial. On the run from New York to Boston a forced-draft trial of about two hours' duration was held. The official trial over the Cape Ann course took

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place November 23, 1904. The speed required by contract was 22 knots per hour.

PRINCIPAL DIMENSIONS OF HULL.

Length between perpendiculars, feet		502
on L.W.L., feet		502
over all, feet		504.04
Beam extreme, feet		69.54
at L.W.L., feet		69.54
Ratio of length to beam		7.22
Depth, molded, main deck, feet		41
Draught, normal, mean, feet		24.08
mean, on trial, feet		24.14
Displacement, normal draught, tons	13,780	
trial mean, tons	13,817	
per inch at L.W.L., normal draught, tons		57
Area of immersed midship section, normal draught, square feet,	1,595	
L.W.L. plane, normal draught, square feet	23,940	
Wetted surface, square feet	43,011	
Center of gravity of L.W.L. plane, aft of M.S., feet		6.56
buoyancy above bottom of keel, feet		13.18
aft of M.S., feet		1.03
Transverse metacenter above center of buoyancy, feet		14.40
Longitudinal metacenter above center of buoyancy, feet		627.00
Coefficient of fineness, block		.5753
of M.S.		.9574
of L.W.L.		.6858
Cylindrical coefficients, longitudinal		.6009
vertical		.8389
Number of frames		124

MAIN BATTERY.

Four 8-inch B. L. R.'s, 45 calibers in length, in two balanced, electrically-operated-turrets.

Fourteen 6-inch R. F. guns, 50 calibers in length; ten on gun deck and four on main deck.

Eighteen 3-inch R. F. guns, 50 calibers in length; eight on gun deck, eight on main deck and two on boat deck.

SECONDARY BATTERY.

Twelve 3-pounders, semi-automatic; ten on boat deck and two on forward bridge.

Four 1-pounders, heavy, semi-automatic in lower tops.

Four 1-pounders, heavy, R. F. guns in upper tops.

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Two 3-inch field guns on carriages.

Two machine guns, 30-caliber on top of turrets.

Six automatic guns, 30-caliber on forward and after bridges.

Two 18-inch submerged torpedo tubes (to be installed), will be located forward of the firerooms.

ARMOR.

Krupp steel for 5 inches and over, the rest is Harveyized.

Belt, 6 and 5 inches at top and bottom, respectively.

Protective deck, 1½ and 4 inches on top and slope, respectively.

Side, above belt, 5 inches.

Athwartship, 4 inches.

Turret gun, 6½ and 6 inches.

Barbettes, 6 and 4 inches.

6-inch gun positions, 5 inches.

2½-inch splinter bulkheads on gun deck.

⅞-inch splinter bulkheads on main deck.

MAIN ENGINES.

There are two main engines, each in a watertight compartment. They are outboard-turning, inverted, direct-acting, 4-cylinder, triple-expansion engines of the contractors' design. The arrangement of cylinders is the H.P. forward; aft of it is the M.P., F.L.P., and A.L.P., respectively, and are connected to prevent athwartship motion only.

The crank shaft is arranged with the H.P. and M.P., also F.L.P. and A.L.P., cranks 180 degrees apart. This makes the sequence of cranks H.P., F.L.P., M.P. and A.L.P. with a 90 degree interval. The crank shaft is in four sections and designed so that the H.P. and M.P. sections are interchangeable, and also the F.L.P. and A.L.P. sections.

The cylinders are supported on inverted Y cast-steel housings on the inboard side, and forged-steel columns outboard. Diagonal braces extend from the bottom of the columns to the top of the housings. There are two braces per housing. Longitudinal braces extend through the steel columns, and tie rods

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connect the tops of columns and housings between cylinders. On the inboard side of the housings is located the reverse shaft. The turning engine is also located on the inboard side of the engines and the throttle valve on the outboard side. All handling gear is located between the H.P. and M.P. housings, and the engine-gauge board is directly back of it on the centerline bulkhead. Jacket steam passes through the top, sides and bottoms, successively, of each cylinder. It is admitted through a reducing valve to the top of the H.P. cylinder and exhausted through a second reducing valve to the top of the M.P. cylinder. It exhausts from this cylinder through a third reducing valve to the top of the L.P. cylinders. The drain from the L.P. jackets is to a trap which discharges to the filter and feed tanks. The clothing material for all cylinders is composed of 85 per cent. carbonate of magnesia, and 15 per cent. asbestos fiber, lagged with Russia sheet iron.

Piston valves used on all cylinders are operated by the Stephenson double-bar link motion with adjustable cut-off. There are five graduations for the points of cut-off on each reverse-shaft arm and are for the down stroke only.

The material for cylinders is cast iron, fitted with close grained, hard cast-iron piston and valve-chest liners. The steam-jacket space is between the piston liner and cylinder body. The body of the H.P. piston valve is cast iron, and of the M.P. and L.P. valves steel pipe with cast-iron heads. The packing rings, of hard cast iron, are designed to prevent expansion beyond a given diameter, but will permit contraction. The H.P. valve stem is connected to the link block, and the M.P. and L.P. valve stems to a cast-steel crosshead which is connected to the link block. The crosshead guides—also of cast steel—are bolted to the valve chests. The suspension links the reverse shaft and eccentric rods, also valve stems, are high-grade machinery forgings.

Data for Cylinders:	H.P.	M.P.	L.P.
Diameter of cylinders, inches	38½	63½	74
Stroke, inches	48	48	48
Thickness of body, inches	2¼	2	1¾

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	H.P.	M.P.	L.P.
Thickness of liner, inches	$1\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{1}{2}$
jacket space, inches	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$
valve chest, inches	$1\frac{5}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$
Number of studs for cylinder cover	48	60	42
Diameter of studs for cylinder cover, inches	$1\frac{5}{8}$	$1\frac{5}{8}$	$1\frac{5}{8}$
Pitch of studs for cylinder cover, inches	3.5	4.8	7.7
Number of studs for valve-chest cover	28	26	18
Diameter of studs for valve-chest cover, inches	$1\frac{1}{4}$	$1\frac{1}{8}$	$1\frac{1}{8}$
pitch circle, inches	$33\frac{1}{4}$	$32\frac{1}{2}$	$34\frac{1}{2}$
Number of tap bolts for piston liner	22	36	42
Diameter of tap bolts for piston liner, inch	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$
pitch circle for piston liner, inches	$36\frac{1}{2}$	$61\frac{1}{2}$	72
Number of piston valves	1	2	2
Diameter of piston valves, inches	26	26	28
balance pistons, inches	6	6	10
Port area through valve-chest liner, square inches	209	228	273
Valve travel, inches	$10\frac{1}{2}$	$10\frac{1}{2}$	$10\frac{1}{2}$
Taken from Port Engine:			
Steam lead, top, inch	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{7}{8}$
bottom, inches	$1\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{3}{16}$
First cut-off, down stroke, per cent	80	77	78
Second cut-off, down stroke, per cent	70	63	67
Third cut-off, down stroke, per cent	57	45	51
First cut-off, up stroke, per cent	77	71	72
Diameter of valve stem through gland, inches	$3\frac{1}{2}$	$3\frac{1}{2}$	$3\frac{1}{2}$
valve, inches	2	2	2
Area of valve-stem crosshead (each side), square inches	27	66	66

Pistons.—The pistons are conical shaped and made of cast steel, finished all over. The upper end is threaded for a nut to be used when removing the piston from its rod. Cast steel is also used for the follower ring, which is held in place by square-necked studs.

The packing rings are of hard cast iron, designed to prevent expansion beyond a given diameter, but allow lineal expansion.

Number of packing rings per piston	2
Width of packing rings, bearing surface, inches	$1\frac{1}{4}$
Thickness of packing rings, inches	1
Number of follower studs, H.P	12
M.P	20
L.P	20
Diameter of follower studs, inches	$1\frac{1}{4}$

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Piston Rods, Crossheads and Crosshead Slippers.—The piston rods and crossheads are high-grade machinery forgings and the crosshead slippers of Parson's manganese-bronze, faced with white metal. The ends of the piston rods are tapered to fit in the piston and crosshead. Crosshead pins on each end of the crosshead are bored out for oil reservoirs. The crosshead is bolted to the slipper. The crosshead guides are of cast iron, made hollow for the water service, and the backing guides are of the same material. They are bolted to the housings.

	H.P. & M.P.	L.P.
Diameter of piston rod, inches	8½	8½
piston-rod hole, inches	4½	5
crosshead pin, inches	11½	9½
Length of crosshead pin (each), inches	12½	10½
slipper, inches	34	34
Width of crosshead slipper, inches	27	18
Length of backing surface, inches	34	34
Width of backing surface (each), inches	9½	6½

Connecting Rods, Crank Shafts and Bed Plates.—The connecting rods and crank shafts are high-grade machinery forgings The upper ends of the connecting rods are forked for the crosshead-pin brasses. These brasses and the crank-pin brasses are lined with white metal. All rods and the crank shafts are hollow. The hollow crank pins are used for oil reservoirs.

The bed plates and main bearing caps are cast steel. The caps and bottom brasses are hollow for water service and lined with white metal. The bottom brasses are circular to facilitate removal.

	H.P. & M.P.	L.P.
Length of connecting rod, center to center, inches	96	96
Diameter of connecting rod, top, inches	7½	6¼
bottom, inches	8¾	7¾
holes, inches	2	2
crosshead-pin bolts, inches	3¾	3
Number of crosshead-pin bolts, per rod	4	4
crank-pin bolts, per rod	2	2
Diameter of crank-pin bolts, inches	5	4
Length of crank pin, inches	22½	15
Diameter of crank pin, inches	20	

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	H.P. & M.P.	L.P.
Diameter of crank-pin hole, inches	12	
crank shaft, inches	18½	
crank-shaft hole, inches	10	
coupling flanges, inches	32½	
Width of coupling flanges, inches	4¼	
Number of coupling bolts	8	
Diameter of coupling bolts, inches	3¾	
pitch circle, inches	26	
Width of crank webs, inches	21¼	
Thickness of crank webs, inches	11½	
Number of bearings	8	
Length of bearings, H.P. and M.P., inches	23	
F.L.P., inches	15½	
A.L.P., inches	15½ and 18½	
	H.P. & M. P.	L.P.
Diameter of main bearing bolts, inches	4⅝	3⅝
Number of main bearing bolts, per bearing	2	2

Reversing Engine.—The reversing engine is located between the L.P. cylinders with its steam cylinder bolted to the outboard side of the main engine bedplate. There is an oil-control cylinder of brass connected to the steam cylinder by tie rods, which serve as crosshead guides. Braces from the housings to the oil cylinder hold it firmly in place. One piston rod extends through each cylinder and is made of forged steel. The tail rod in the oil cylinder is brass. A hand oil pump, located near the handling gear of the main engines, is connected to the oil cylinder. The reverse shaft is near the top of the inboard side of the housings, and the levers are connected to the crosshead of the reverse engine by two forged-steel connecting rods. Differential levers control the valve mechanism.

Data for Engine:

Diameter of steam cylinder, inches	16
oil cylinder, inches	10
steam piston rod, inches	3⅝
oil piston rod, inches	3
tail rod, inches	3
Length of crosshead bearing, inches	10
Width of crosshead bearing each side, inches	2½
Length of crosshead pin, each, inches	3½
Diameter of crosshead pin, each, inches	3½

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	H.P. & M.P.	L.P.
Diameter of tie rods between cylinders at crosshead bearing, inches...		3
ends, inches		$2\frac{3}{8}$
connecting rods, end, inches		$2\frac{1}{2}$
middle, inches		$2\frac{7}{8}$
Length of connecting rods, feet and inches	7–	$2\frac{3}{4}$
reverse levers, inches		$17\frac{5}{8}$
Diameter of reverse shaft, inches		$8\frac{1}{2}$
bearings, inches	$8\frac{1}{2}$,	7, 5
Length of bearings, inches	$8\frac{1}{2}$,	$8\frac{1}{2}$, $8\frac{1}{2}$

Turning Engine.—The turning engine is located on the forward side of the F.L.P. housing with the crank shaft horizontal. The shaft extends outboard to a roller bearing bolted on a steel column; for turning the main engines by hand the end of this shaft is made square to fit a wrench. The worm on the crank shaft engages a wheel at the top of a vertical shaft, and on this shaft is a second worm, prevented from turning by a feather but may be moved vertically to engage or disengage with the wheel on the main crank shaft. The step bearing for this vertical shaft is on the engine foundations. A plug cock designed to reverse the direction of the steam and exhaust is used to reverse the engine.

The time required to turn the main-engine crank shaft through one revolution is about three and one-half minutes.

Data for Turning Gear:

Number of cylinders	2
Diameter of cylinder, inches	8
piston rod, inches	$1\frac{1}{2}$
valve, inches	$2\frac{1}{2}$
Stroke, inches	8
Valve travel, inches	$1\frac{1}{2}$
Crank angle, degrees	90
Area of crosshead slipper, square inches	23
Diameter of crosshead pin, inches	2
Length of crosshead pin, inches	$2\frac{1}{2}$
Diameter of crank pin, inches	$2\frac{1}{2}$
Length of crank pin, inches	$3\frac{5}{8}$
Diameter of crank-shaft, inches	$2\frac{1}{2}$
Length of crank-shaft bearings, inches	4, 6
Diameter of pitch circle of worm, inches	$4\frac{1}{2}$
wheel, inches	$14\frac{1}{2}$
Number of teeth	30

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Diameter of pitch circle of vertical worm, inches	9.68
wheel, inches	57.30
Number of teeth	60
Diameter of vertical shaft, inches	4
Area of step bearings, square inches	26

Shafting and Thrust Bearing.—The shafting is installed in four sections—thrust, line, stern-tube and propeller—of high-grade machinery forged steel, oil tempered. The line and stern-tube shaft coupling is made as follows: A forged-steel ring fits on the end of the stern-tube shaft—four keys prevent it rotating—and into a circumferential groove in the shaft are fitted two half rings. The coupling bolts go through the steel ring, the half rings and the flange of the line shaft. Into the outboard coupling sleeve the shafts are tapered and are held in place by longitudinal and through keys. The stern-tube shaft is encased in composition casings and the space between them and the shaft is filled with rosin and tallow.

The thrust bearing is of the horseshoe type. The body is of cast iron, so shaped that it forms a reservoir for oil. The shaft bearings at the ends are fitted with glands to prevent the leakage of oil. To the foundations is riveted the sole plate, and the thrust-bearing body is bolted to this, there being steel wedges at each end to provide for longitudinal adjustment.

The thrust shoes are cast steel, hollow for water service, and faced with white metal. At the top is an oil reservoir and from it are oil-ways to the bearing faces.

Data for Shafting:

Diameter of thrust shaft, inches	17½
hole, inches	10
Number of thrust-shaft collars	13
Outside diameter of thrust-shaft collars, inches	28
Width of thrust-shaft collars, inches	2
space of thrust-shaft collars, inches	4½
Length of thrust-shaft bearing, inches	18½
Number of thrust shoes	12
Diameter of thrust side rods, inches	4
line shaft, inches	17½
Diameter of line shaft hole, inches	10½
Length of steady bearing, inches	24
Diameter of stern-tube shaft, inches	18½

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Diameter of stern-tube shaft hole, inches	10½
Length of stern-tube forward bearing, inches	41⅝
after bearing, inches	54½
Diameter of propeller shaft, inches	18½
hole, inches	10½
Length of forward strut bearing, inches	30
after strut bearing, inches	57⅝
Diameter of coupling flange, inches	32½
Thickness of coupling flange, inches	4¼
Number of coupling bolts	8
Diameter of coupling bolts, inches	3¾
Diameter of coupling bolts, pitch circle, inches	26
Length of outboard coupling sleeve, inches	72
Outside diameter of outboard coupling sleeve, inches	24
Width of longitudinal keys, inches	2½
Depth of longitudinal keys, inches	1½
Width of through keys, top, inches	8
bottom, inches	7
Thickness of through keys, inches	2½

Propellers.—The propeller blades and hubs are manganese-bronze. The hub is fitted on the tapered end of the propeller shaft and held in place by one longitudinal key and a nut on end of shaft threaded in the opposite direction to the rotation of the propeller. The blades fit in recesses in the hub and are held in place by rolled manganese tap bolts—a small casting fitting between the bolts prevents their turning. The pitch is adustable by making the bolt holes elongated, and is variable, due to the working surface being made slightly convex. The propeller was carefully balanced, file finished all over, and tinned.

The starboard propeller rotates in a clockwise direction, and the port in a counter clockwise direction.

Data for One Propeller:

Number of blades	3
Diameter of propeller, feet and inches	18– 0
Pitch for official trial, feet and inches	22– 0
maximum, feet and inches	23– 6
minimum, feet and inches	20– 6
Ratio of pitch to diameter	1.22
Helicoidal area, square feet	92
Projected area, square feet	74
Disc area, square feet	254
Ratio projected to disc area	.29

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Area of immersed midship-section, square feet	1,595
Ratio disc to immersed midship-section area	.1595
projected to immersed midship-section area	.0454
Number of tap bolts for blades	27
Diameter of tap bolts for blades, inches	4
Length of tap bolts for blades, under head, inches	7⅞
Width of longitudinal key, inches	4¼
Thickness of longitudinal key, inches	2¼
Length of longitudinal key, inches	40
Diameter of nut thread, inches	12
Length of nut thread, inches	10
Threads per inch	4
Immersion of upper tip of blade, inches	66
Height of lower tip above keel, inches	12

Data from Official Trial:	*Port.*	*Stbd.*
Average revolutions per minute while on course	128.5	128.4
Slip in per cent. of its own speed	19.42	19.36

Water and Oil Services.—The water-service pipe is connected to a nozzle on the main circulating pump discharge to the main condenser. This pipe leads below the floor plates to the inboard side of the main engine and has extensions to distributing manifolds placed in the housings. These manifolds are inverted brass V-castings with distributing valves to—

1 Crank pin.
2 Crank-pin oil reservoir.
3 Crosshead-pin oil reservoirs, guides and slipper.
4 Crossheads.
5 Main bearings.
6 Main bearing caps.

There is an extension from the water-service pipe to the thrust bearing, steady bearing, and stern-tube stuffing box.

On the after athwartship bulkhead is placed a small pump to supply oil to the distributing tank—on the center-line bulkhead below the berth deck.

A pipe from this tank has extensions to all the oil boxes on the main and auxiliary engines and to oil cans in the engine room. All oil boxes are arranged for sight and wick feeds. The oil pipe to the crosshead pin is connected to the water-

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service pipe through a T-fitting, thus permitting water to be used with the oil, and causing it to saponify.

The crank pins are hollow, and this volume is used for an oil reservoir. Banjo oilers are fitted to transmit oil and the water service to these reservoirs.

Indicator Cards.—One set of cards from the port engine, representing average conditions, were selected for comparison with theoretical cards. These results are subject to a small correction due to the steam being admitted to the M.P. receivers during most of the trial. Card No. 4 was selected for this purpose and the comparison with the average is given below:

Port Engine.

	Card No. 4. M.E.P.	Average from trial. M.E.P.
H.P. cylinder	114.50	112.64
I.P. cylinder	60.75	59.43
F.L.P. cylinder	17.65	17.39
A.L.P. cylinder	17.40	17.32

Starboard Engine.

H.P. cylinder	116.00	109.86
I.P. cylinder	57.50	55.74
F.L.P. cylinder	16.00	17.02
A.L.P. cylinder	17.00	16.04

	H.P.	I.P.	Combined F.L.P.,	A.L.P.
Area of piston, top, square inches	1,164	3,167	8,302	
Ratio of areas	1	2.72	7.38	
Per cent. cut off, top	80	77	78	
bottom	77	71	72	
clearance, top	27.8	25.4	25.1	26
bottom	28.1	17.1	17.9	18

Volume at cut-off, top, H.P. cylinder, cubic inches	44,800
of clearance, top, H.P. cylinder, cubic inches	15,400
at cut-off, total, H.P. cylinder, cubic inches	60,200
bottom, H.P. cylinder, cubic inches	40,900
of clearance, bottom, H.P. cylinder, cubic inches	14,900
at cut-off, total, H.P. cylinder, cubic inches	55,800
Mean of top and bottom volumes, H.P. cylinder, cubic inches	58,000
Piston displacement + clearance, top, L.P. cylinder, cubic inches,	518,400
bottom, L.P. cylinder, cu. inches,	480,600
mean, L.P. cylinder, cu. inches,	499,500
Ratio of initial to final volumes of steam	8.62
Mean referred pressure from trial data, pounds	51.9
theoretical, pounds	91.6
Ratio of actual to theoretical	.567

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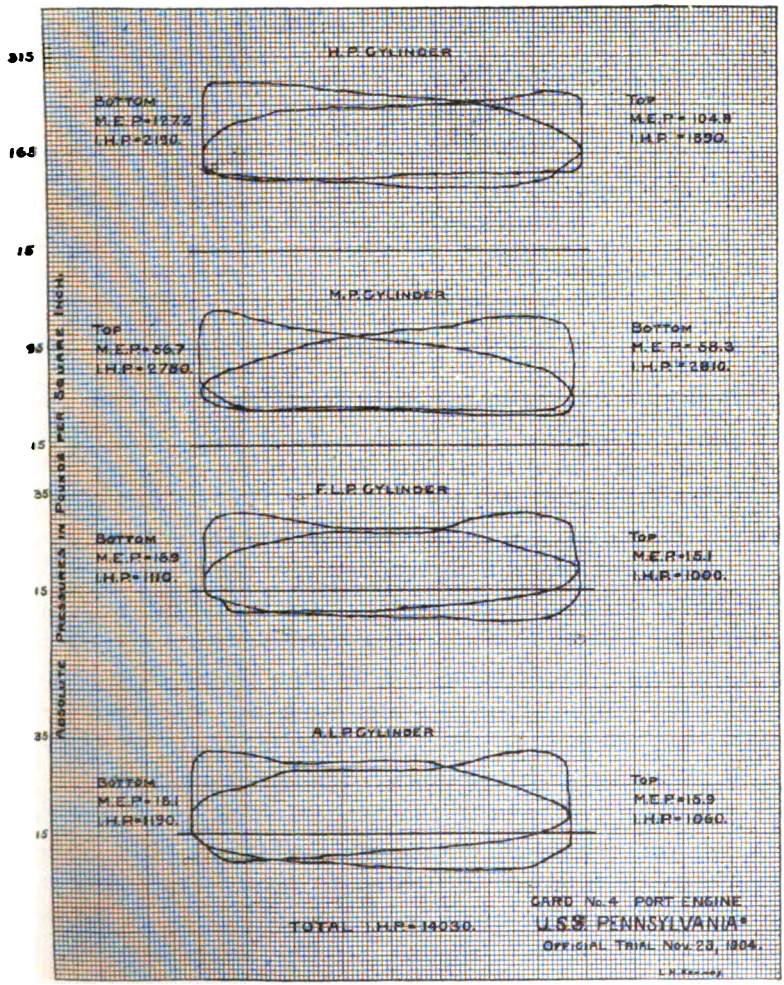

INDICATOR CARDS.

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BOILERS.

The Niclausse boilers installed on this ship were designed by the contractors and built by the Stirling Boiler Company. The arrangement on board ship is symmetrical with respect to the center-line bulkhead and is as follows:

Forward Compartment:

1 Battery of 4 boilers.
1 Battery of 2 boilers.

Middle Compartment:

2 Batteries of 2 boilers each.

After Compartment:

1 Battery of 2 boilers.
1 Battery of 4 boilers.

The drums of these boilers are short and permit of two being placed athwartship. The heads of the drums are spherical, placed in one end convex and in the other concave. The concave ends of each pair are together. These drums are outside of the casings and do not come in contact with the heat. Below the drums are the headers, and into these are fitted the circulating and generating tubes. These tubes are supported at the other end by cast-iron plates with holes to correspond to the arrangement of tubes. A division plate in the header divides the down-current of water from the up-current containing the steam. In the generating tubes are fitted concentrically the circulating tubes. The direction of the flow of water is down the front of the headers—this side is not exposed to heat—to the circulating tubes, which are fitted closely into the division plates, and through them to the end of the generating tube, from which point it flows back to the headers and up to the drum. Tapped into the lower end of each header is a pipe to a bottom blow casting.

The baffling is obtained by placing between generating tubes steel tubes of the same length. This constrains the smoke to move athwartship but continuously upward.

The casing for the boilers is made of steel plates, stiffened

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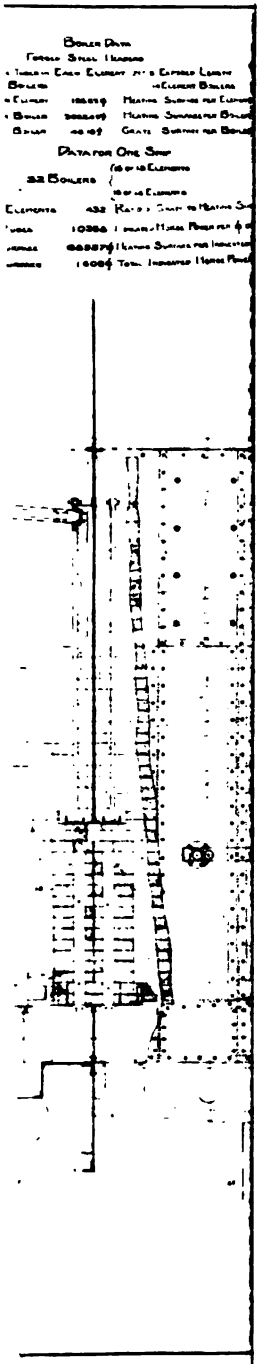

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by angles. The side, back and front casings are vertical and rectangular. Fire brick and magnesia are the materials used for insulating; fire brick extends to the top of the side casings and to the tubes on the back and division casings. The furnace extends the full width of the boiler and the length of the tubes. A shaking grate of the contractors' design is installed. In the battery of two boilers there is a division wall of brick between furnaces, and in the battery of four boilers the furnaces are divided in the same manner. For each furnace there are three furnace doors, which swing on an horizontal axis, with a counterweight added to balance them, and below are three ash-pit doors similarly arranged.

A dry pipe is placed in the top of each drum, terminating in a main steam casting, also on top of drum. To this casting are bolted the safety valve and main stop valve, designed also for an emergency stop valve. The safety valve may be operated, by lifting gear, from the fireroom floor or gun deck. A sentinel valve and air cock are also on top of the drum. The feed valves are on the side of the drum, and the main and auxiliary feed-check valves are in one casting. The auxiliary feed is admitted to the feed-check valve through a valve regulated by an extension spindle from the fireroom floor. The main feed-regulating valve is operated by a float attached to a lever, and this lever is attached to the spindle of a rotating valve. The main feed passes through this valve, thence through the feed-check valve to the boiler. To the spindle of this regulating valve is attached a short lever, and from it hangs a rod to be used for operating the valve from the fireroom floor in case of emergency. The addition of weights to this rod decreases the immersion of the float and permits of changing the steaming water level. Two water-gauge columns are installed; one is the ordinary cylindrical glass and the other the Klinger reflex water gauge. These are on each side of a composition casting connected at the lower end to a valve on the drum and at the upper end by a short piece of copper pipe to a second valve on the drum. There are also three try cocks for testing the water level. The cylindrical

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glasses are fitted with quick-closing cocks operated from the fire-room floor, and reflex glasses by automatic closing cocks. The internal fittings of the drum consist of a division plate for the down and up-currents of water; this extends through the nipples to the division plates in the headers. On the division plate are placed the zincs. To prevent wide variations for the float of the feed-regulating valve, due to the ship rolling, it is partially encased in a compartment perforated at the bottom. A scum pan for the surface blow is also fitted. All rivet holes were drilled through plates held together by tack bolts. The tack-bolt holes were $\frac{1}{8}$-inch smaller than the rivet and reamed to size. The weight of headers and drum is borne by the furnace front casings.

Boiler Drums.—The shell is made of one steel plate rolled to the required diameter, and its edges do not meet except at the ends where they are riveted to the heads. The shell plate is treble-riveted between inside and outside butt straps of the same width. The outside butt strap is flanged outward from the center of the boiler and machined to receive the cone nipples from the headers, while the inside butt strap is flanged in the opposite direction. This flanging is done by forcing through the plates a circular die. The manhole is cut through the shell, and on the inside is riveted a reinforcing plate flanged inward. The heads are spherical shaped with a radius equal to the internal diameter of the drum.

Headers.—The headers are forged from seamless-drawn steel tubes. The upper end is drawn down and expanded into a flange and machined to receive the cone nipple from the drum. This cone nipple is also of steel tubing machined to taper towards each end and fit accurately into the header and drum. Four bolts connect the header and drum and are inserted from inside of the drum. The head of the bolt is conical, fitting accurately into a conical hole in the outer butt strap. The other forging processes are to square the tube; insert a cast-iron mandrel and forge to the sinuous form—the cast-iron mandrel is then broken and removed—holes for the tubes are drilled and flanged outward; a steel plate is welded into the

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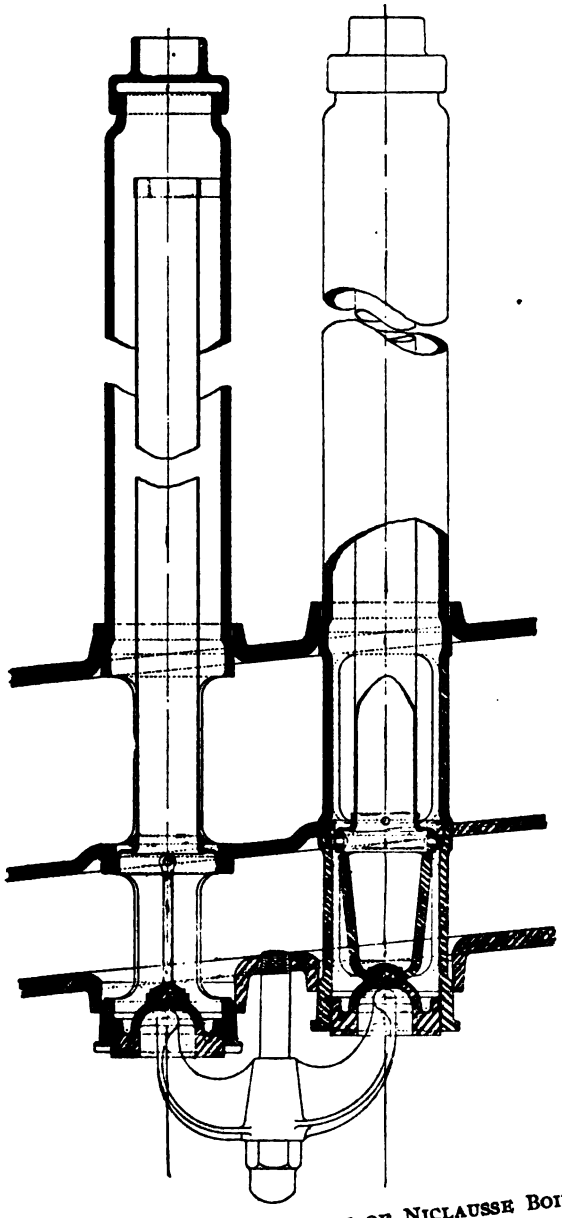

ARRANGEMENT OF TUBES IN HEADERS OF NICLAUSSE BOILER. UPPER TUBE SHOWN IN ELEVATION. LOWER TUBE SHOWN IN PLAN.

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bottom and the division plate, machined to correspond to the sinuous form, is introduced cold into the hot header. The headers are then machined for the tubes.

The Tubes.—The generating tubes are seamless-drawn steel, and forged down at one end for a drop-forged nut in order that it may go through the header. The lantern end of the tube is upset—for threads to fit the circulating tube plug—and expanded in three places to make the joints on each side of the header and to fit closely into the division plate. The tubes are machined to fit accurately into the cone bearings of the headers, and milled out between the bearings for the circulation of water. The circulating tube is made of thin sheet steel attached to a plug which screws into the generating tube. These tubes are held in place by a drop-forged steel dog for each pair.

Installation of Boilers.—The boilers were installed in the following manner: The back casing is put in place followed by the sides, furnace fronts and division wall and filled with magnesia. The brick work is next installed, and after this the headers with the tubes assembled in them. At this time the baffling tubes are inserted. The drum is then connected to the headers and the boiler subjected to a hydraulic test of 450 pounds by gauge. The casing is then completed on top, for the uptakes, and the cleaning doors in front installed. To clean the tubes these doors are opened and a steam lance inserted between the headers. The clothing for the boiler drums is magnesia lagged with galvanized steel plates.

On board ship the 13-header boilers are inboard and the 14-header outboard.

Data for Boilers:	*13 header.*	*14 header.*
Number of boilers	16	16
Length of drum, feet and inches	8– 9¼	9– 4⅞
Outside diameter of drum, inches	42	42
Thickness of drum plate, inch	¾	¾
heads, convex, inch	¾	¾
concave, inches	1	1
Efficiency of plate section, per cent	76.80	76.29
rivet section, per cent	111.17	110.59
Heating surface per header and tubes, square feet	158.7	158.7

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	13 header.	*14 header.*
Total heating surface, square feet	2,062	2,221
grate surface, square feet	48	52
Ratio H.S. to G.S.	42.97	42.71
Total heating surface for all boilers, square feet	68,537	
grate surface for all boilers, square feet	1,600	
Ratio total H.S. to G.S.	42.84	
Number of boilers per smoke pipe	8	
Height of smoke pipe above base line, feet	100	
Area through smoke pipe, square feet	68.4	
Ratio G.S. to stack area	5.85	
Diameter of twin safety valve, inches	$2\frac{1}{2}$	
Number of generating tubes per header	24	
Length of generating tubes, feet and inches	8– $1\frac{1}{2}$	
Thickness of generating tubes, B.W.G.	6	
Diameter of generating tubes, outside, inches	$3\frac{1}{2}$	
Length of circulating tube, feet and inches	8– 0	
Diameter of circulating tube, outside, inches	$1\frac{9}{16}$	
Thickness of circulating tube, B.W.G.	21	
Area for water through nipple, square inches	8.6	
header, square inches	14.9	
lantern, square inches	7.6	
circulating tube, square inches	1.77	
steam and water through annular space in generating tube, square inches	4.41	
steam and water through lantern, square inches	10.1	
header, square inches	26.7	
nipple, square inches	12.0	
Inclination of tubes	1 to 10	
Length over all of firerooms in ship, feet	128	
Width over all of firerooms in ship, feet	23	
Total area of floors of firerooms in ship, square feet	5,888	
Total area of floor space occupied by boilers, square feet	2,570	
Ratio of boiler F.S. to fireroom floor area	.436	
Total volume occupied by boilers, cubic feet	34,600	
of firerooms above floor plates, cubic feet	122,800	
Ratio of boiler to fireroom volume	.282	
Weight of boiler, dry, per cubic feet, pounds	53.09	
water in 13-header boiler, pounds	7,070	
14-header boiler, pounds	7,610	
Total weight of water in all boilers, pounds	234,880	
Weight of boiler, wet, per cubic foot	62.91	
Ratio of weight of boilers dry to boilers wet	.88	

Data from Official Trial:

Thickness of fires, average, inches	6 to 8	
Pounds of coal burned per hour	61,546	
per square foot of grate per hour		38.46

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	13 header.	*14 header.*
Pounds of coal burned per square foot of H.S., per hour..		.898
I.H.P. of main engines per hour.....		2.18
total I.H.P. of main and auxiliary engines, per hour.....		2.12
I.H.P. per square foot G.S., main engines.....		17.5
and auxiliaries...		18.1
H.S., main engines and auxiliaries...		.42
H.S., main engines.....		.41
Steam pressure, gauge, average, pounds.....		300.2
Average temperature at base of smoke pipe, degrees Fahrenheit.....		630
fireroom floors, degrees Fahrenheit.....		82.8
fireroom gratings, degrees Fahrenheit.....		89.7
of atmosphere, degrees Fahrenheit...		59.5
draft at base of smoke pipe, in inches of water...		—.34
air pressure in firerooms, in inches of water.....		1.17

STEAM AND EXHAUST PIPING.

The auxiliary steam pipes, one port and one starboard, are suspended from the berth deck. They are parallel to the center-line bulkhead, and extend from a distributing casting on the engine-room bulkhead to the forward fire rooms, and from here are extensions to the dynamos, to the ice machine and anchor-hoist engine through a reducing valve on the port side of the upper dynamo room. A nozzle is provided in the dynamo room for steam to forward heating system and to galley. In the firerooms are extensions to pumps, to blowers and from them to the ash-hoist engines, to fire extinguishers in bunkers, and on port side to sea chests; nozzles are provided for boiler-tube cleaners, one forward for steam to dynamo condenser pump and one on starboard side for steam to whistle. Through a reducing valve on the starboard side in the machine shop steam is supplied the evaporating plant.

The distributing casting in the engine room has three nozzles: one for a cross-connecting pipe which is connected to the main steam through a cross valve at the separator, and from this valve are extensions to the fire and bilge pump and main injection sea chest—from the separator is a bleeder to main

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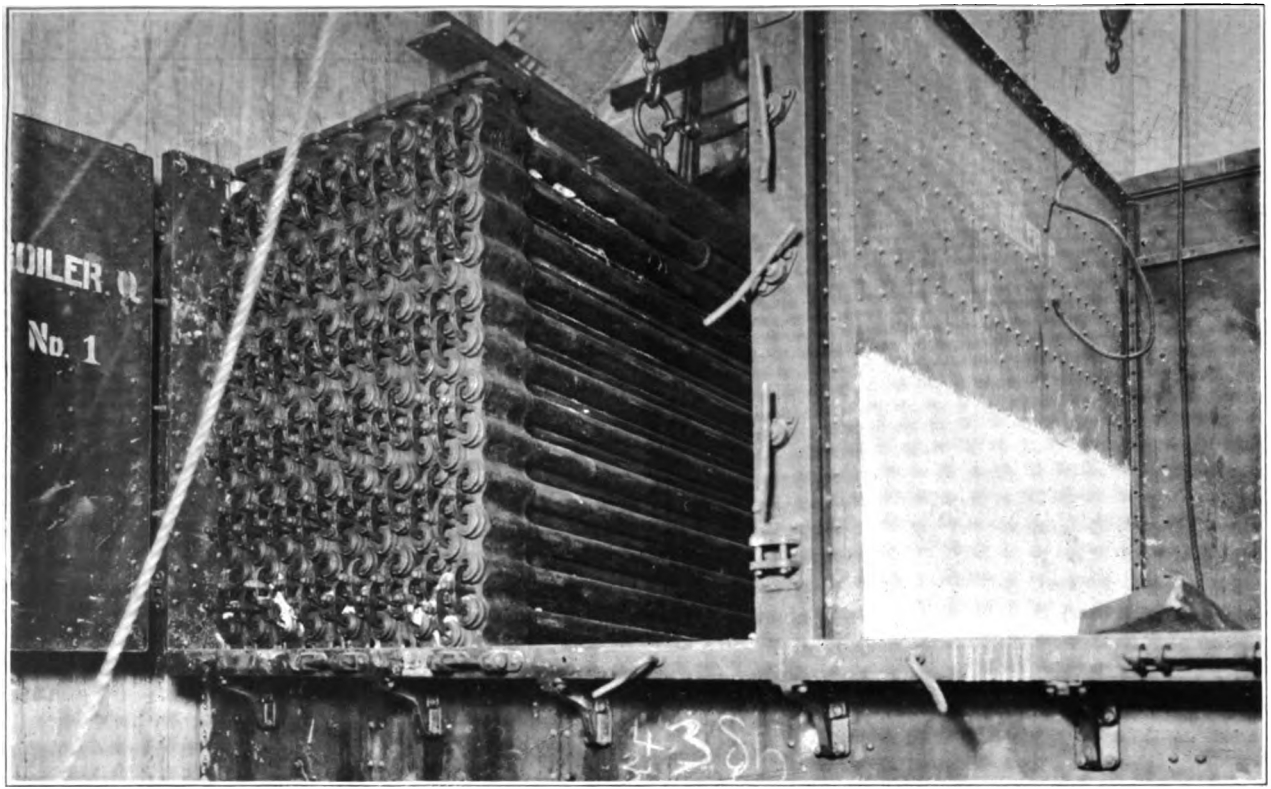

BOILER OF U. S. S. "PENNSYLVANIA."

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condenser; the second supplies steam to the main circulating pump engine, and from this pipe is an extension to a manifold which distributes steam to the hot-well pump, to auxiliary sea chest and for boiling out condenser; the third supplies steam to the main air pump and to another distributing casting in the after end of the engine room. The latter casting distributes steam to turning engine, reversing engine, auxiliary condenser engine and on starboard side to after heating system and to steering engine through a separator located near the engine.

Steam is supplied to the auxiliary steam pipes from each pair of boilers through an angle valve bolted to a nozzle on these pipes, and on the bottom of this valve is bolted a sleeve fitting into an expansion joint at the top of an inverted T-casting. This casting rests on one of the boiler drums, and on each side are expansion joints for the steam pipes from the main steam casting of the two boilers.

The main steam pipe and auxiliary steam pipe are the same for the four forward boilers, but aft of them these pipes are separate and parallel. Steam is supplied the main steam pipe through three cross-connecting valves on the auxiliary steam pipe; one valve is located in each boiler compartment. In the starboard fireroom are extensions from the main steam pipe to the main feed pumps. On the forward bulkhead of the engine room is the main steam stop valve, which may be operated from the gun deck. Below the stop valve is the separator, and connecting this with the throttle is a short piece of pipe with an expansion joint at each end. The diameter of this pipe is $14\frac{7}{8}$ inches. Between the stop valves is a cross-connecting pipe, and an extension from this supplies steam to the M.P. and L.P. receivers.

The boiler stop valve is designed to be closed by steam pressure acting on a piston on the valve stem. Steam is applied to the piston through a valve on the opposite side of the center-line bulkhead, and is operated, by an extension spindle, from the fireroom floor. There is an extension from the auxiliary steam to these bulkhead valves.

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HEATING SYSTEM.

The heating system is divided into two sections, one forward and one aft. The method for each section is to supply steam through a reducing valve to a distributing manifold. To this manifold are connected a relief valve and a steam gauge. This manifold admits steam to the several circuits, which drain to a second manifold, and through a trap to the auxiliary exhaust. The manifolds for the forward sections are in the upper port dynamo room, and for the after sections in the after end of the starboard engine room. There is an independent circuit for the steam to pantries, officers' shower baths, and heaters in their bath rooms, which also drains, through a trap, to the auxiliary exhaust.

STEAM PIPE DRAINS.

The drains from the steam pipes in the forward boiler compartments are led to a trap which discharges to the main feed tank. This trap is located on the outboard bulkhead of the firerooms on a level with the gratings, and the trap-discharge pipe is supported from this bulkhead. The discharge from the dynamo trap is to this pipe. There is one trap located as above in each boiler compartment and each is arranged to be bypassed.

In the engine room the drains from the auxiliary steam distributing casting and steam to receivers at engine discharge through a trap to the trap discharge pipe. The trap of the main steam separator and the traps of the evaporator drains also discharge into this pipe. The feed-water heater drains either to the main or auxiliary condenser. The cylinder and valve chests drain to the main condenser and the jackets to a trap which discharges to the feed tank. All drains of auxiliary cylinders are led to the auxiliary exhaust and to bilge.

EXHAUST PIPING.

The main exhaust pipes are from the L.P. cylinders to the main condenser. The auxiliary exhaust pipes are parallel to the auxiliary steam pipes through the firerooms. They are

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cross connected through the dynamo rooms where they are connected to the dynamo exhaust and exhaust from the ice machine and anchor-hoist engine. In the firerooms they receive the exhaust from pumps, ash-hoist engines and blowers, and in the after firerooms are extensions to the escape pipe on after stack. A nozzle is provided in the machine shop for vapor from the evaporators.

The auxiliary exhaust in the engine rooms leads to the main and auxiliary condensers through spring relief valves. These valves are provided to produce a back pressure for the feed-water heater or L.P. receivers. At the forward end of the engine room a nozzle on the bottom of the auxiliary exhaust is for the connection to the feed-water heater. This nozzle acts as a drain for the exhaust from forward. On the port side is the exhaust from steering engine. The exhaust from the main air pump is to the main condenser or L.P. receivers, and the auxiliary exhaust may also be sent into the L.P. receivers. The joints for the steam and exhaust piping are made with corrugated copper gaskets and red lead putty. Plain flanges are used, file finished from the bolt circle towards the center of pipe and tested with a face plate; on this surface the joint is made and the remainder of the flange is filed a little lower. All pipes were tested after installation with 325 pounds of steam, and found generally to be tight, and no leaks have since developed.

The material used for flanges on all work under the cognizance of the Bureau of Steam Engineering is as follows: composition flanges on all pipes below floor plates, drop-forged steel flanges for all pipes above floor plates of 5½ inches in diameter and smaller; for larger pipes cast-steel was used except for the main and auxiliary steam lines, which are steel pipes with welded flanges. Rubber gaskets were used for all water piping.

Auxiliary Condenser.—There is one auxiliary condenser in each engine room. The tubes are vertical, and on one side of condenser is bolted the circulating engine. The impeller is on the end of the crank shaft, and rods extending down from

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the crosshead are connected to the air pump. The circulating water is on the outside of the tubes.

Data for one Condenser:

Number of tubes	1,161
B.W.G. of tubes	16
Diameter of tubes, inch	$\frac{5}{8}$
Length of tubes, inches	$52\frac{1}{2}$
Cooling surface, square feet	759
Diameter of steam cylinder, inches	$6\frac{3}{4}$
piston rod, inches	$1\frac{5}{8}$
Stroke, inches	7
Length of connecting rod, center to center, inches	14
Valve travel, inches	$1\frac{5}{8}$
Diameter of crosshead pin, inches	$1\frac{1}{2}$
Length of crosshead pin, inches	$2\frac{3}{4}$
Diameter of crank pin, inches	$2\frac{1}{2}$
Length of crank pin, inches	$2\frac{1}{4}$
Diameter of crank shaft, inches	$2\frac{1}{2}$
Length of bearings, inches	4 and $3\frac{1}{2}$
Area of crosshead slipper, square inches	24
Diameter of air pump piston, inches	$7\frac{1}{2}$
rod, inches	$1\frac{1}{2}$
Diameter of impeller, inches	24
Width of tip of blades, inches	$2\frac{3}{4}$

The air pump valves are rubber.

Main Condenser.—One main condenser is located on the outboard side of each engine room. The shell plating is of steel and the heads of composition. Brass tubes fitting in stuffing boxes in Muntz-metal tube sheets provide the cooling surface. In the forward head is a valve to discharge the injection water directly overboard, and zinc plates are bolted to the manhole covers.

Data for One Condenser:

Number of tubes	5,662
B.W.G. of tubes	18
Diameter of tubes, inch	$\frac{5}{8}$
Length of tubes, feet and inches	15- 8
Number of supporting plates	2
Cooling surface, square feet	14,359

Data from Trial:	*Port.*	*Starboard.*
Temperature of injection water, degrees Fahrenheit	56.6	56.6
discharge water, degrees Fahrenheit	117.2	107.8
Ratio of cooling surface to I.H.P. of main engines	1.05	1.00

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The Trial Board found upon inspection, after the trial, that the condensers were perfectly tight. The packing for the tubes was inserted by a Star Condenser Packing Tool, the patent for which is held by one of the erecting engineers of the contractors. One end of the packing tape is dipped into tallow and inserted in the tube sheet; the remainder of the operation is done by the packing tool. The dry tape when brought in contact with the water will swell, assisting in making a tight joint. The condensers are clothed with hair felt and lagged with teak.

The Main Circulating Pump.—Located forward of the main condenser is the main circulating pump. This pump has three suctions—the main injection, engine-room bilge and the main drain; these suctions are designed so that only one can be open at a time. The casing and impeller of the circulating pump are composition. The capacity of this pump as required by the specifications is 15,000 gallons when making 245 revotions per minute. A test was made to determine the capacity of the pump in the following manner:

The engine-room bilge was flooded to the floor plates and the bilge injection valve opened to the pump which discharged directly overboard. An observation was made on the water level at the beginning, and at the end of the trial thirty seconds later. Several trials were made to determine the average change in water level. The revolutions of the engine were very nearly 245 per minute. The volume of water pumped out may be determined by flooding the bilge from the final to the initial water level by emptying into it a known volume of water. Indicator cards were taken during the trial. To determine whether the engine can maintain the conditions for a given time it will be necessary to connect to the main injection.

Data for Pump:

Diameter of suction nozzle, inches	21
discharge nozzle, inches	21
impeller, inches	54
Width of tip of blades, inches	5¼
Diameter of H.P. piston, inches	11

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Diameter of L.P. piston, inches		19
piston rods, inches		$2\frac{3}{4}$
Stroke, inches		12
Length of connecting rod, center to center, inches		27
Valve travel, inches		$3\frac{1}{8}$
Angular advance H.P., degrees		35
L.P., degrees		$37\frac{1}{2}$
	Top.	*Bottom.*
Steam lead, H.P., inch	$\frac{1}{8}$	$\frac{3}{16}$
L.P., inch	$\frac{3}{16}$	$\frac{1}{4}$
Cut off, H.P	.75	.69
L.P.	.74	.67
Diameter of H.P. valve, inches		4
L.P. valve, inches		9
Diameter of crosshead pin, inches		4
Length of crosshead pin, inches		$5\frac{1}{2}$
Diameter of crank pin, inches		$5\frac{1}{2}$
Length of crank pin, inches		$6\frac{1}{2}$
Diameter of crank shaft, inches		$5\frac{1}{4}$
Lengths of bearings, inches		10, 10, $6\frac{1}{2}$
Crank angle, degrees		90
Area of crosshead slipper, square inches		74

Data from Test:

Revolutions per minute	246
I.H.P.	249
Rate of discharge in gallons per minute	18,000

Data from Official Trial:

	Port	*Starboard.*
Average revolutions per minute	166.5	213.6
Average I.H.P.	92.2	155.9

Main Air Pumps.—Aft of the main condensers are installed the main air pumps. These are twin vertical-beam air pumps of the Blake design. The air pump discharges into the filter and feed tank in after end of engine room.

Data for Pump:

Diameter of steam cylinders, each, inches	14
water cylinders, each, inches	35
Stroke, inches	18
Diameter of steam piston rod, inches	$2\frac{3}{4}$
air piston rod, inches	$3\frac{1}{2}$
suction nozzle, inches	13
discharge nozzle, inches	11

The air-pump valves are phosphor-bronze discs.

Data from Official Trial:

	Port.	*Starboard.*
Average strokes, per minute	22.80	23.10
vacuum, inches	26.00	26.66
indicated horsepower	18.74	17.27

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FEED SYSTEM.

Feed and Filter Tank.—The top of the feed tank is arranged as a filter, and the filtering material is "loofa." This is placed between horizontal perforated plates. The filtering chamber is divided by vertical divisions so that the water flows from the bottom of one to the top of the succeeding one.

Below the filtering chamber is the feed tank. All material is steel plates and angles, and is galvanized. The clothing is hair-felt lagged with galvanized steel plates.

Capacity of each feed and filter tank, cubic feet, 711.

Capacity of reserve feed tanks, inner bottom,

	Cubic feet.
B, 87	634.1
B, 88	634.1
B, 89	619.0
B, 90	619.0
B, 91	589.8
B, 92	589.8
B, 93	942.8
B, 94	942.8
B, 95	942.8
B, 96	942.8
B, 97	937.7
B, 98	937.8
B, 99	2,224.6
	11,557

Hotwell Pumps.—The hotwell pumps are placed beneath the main condensers. They are 12 inches by 16 inches by 16 inches, horizontal, simplex, piston pumps, of Blake design. The suction manifold has the following valves:

1. From either filter and feed tank.
2. From either air-pump channel way.
3. From reserve feed tanks.
4. From water-boat filling pipe.

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The discharge manifold:

1. To filter and feed tank.
2. To reserve feed tanks.
3. To feed system.

Data from Official Trial:

Average strokes of port pump	18.5
starboard pump	18.5
indicated horsepower	18.2

The discharge to the feed system is through a grease extractor and a feed heater, located on the forward athwartship bulkhead, one of each port and starboard. Each of these may be by-passed.

Turkish toweling is used for filtering material in the grease extractors. This is drawn over perforated brass tubes. The feed-water heater has a nozzle on the side to admit the auxiliary exhaust steam, and drains to the main or auxiliary condenser. The feed water is admitted at the bottom, and after passing through the tubes it is discharged at the top to the main feed pump suction.

Data for One Heater:

Number of tubes	832
B. W. G. of tubes	18
Diameter of tubes, inches	$\frac{5}{8}$
Length of tubes, feet and inches	7- $2\frac{1}{4}$
Area of heating surface (tubes), square feet	952
(tube sheet), square feet	3
total, square feet	955

Data from Official Trial:	*Port.*	*Stbd.*
Average temperature of water to feed heater, degrees Fahrenheit	85.7	80.4
from feed heater, deg. Fahrenheit	150.2	148.5

Grease extractor by-passed.

The feed-water pipes from the heaters pass forward into the firerooms and meet on the center-line bulkhead extending forward on the starboard side above the main feed pumps. There are extensions from it down to the four 12-inch by $7\frac{5}{8}$-inch by 12-inch vertical, duplex, center-packed plunger, Blake design, main feed pumps. These pumps discharge to a feed main below and parallel to the feed-supply pipe; from this pipe are extensions to the starboard boilers and through the center-line bulkhead to the port boilers.

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Data from Official Trial:

Average double strokes per minute of all pumps	37.2
Total indicated horsepower	310
Average pressure maintained in feed main, pounds	357

On the port side of the center-line bulkhead opposite the main feed pumps are the auxiliary feed pumps, of the same dimensions and design. The suction manifold has the following valves:

1. Suction from sea.
2. Suction from auxiliary drain.
3. Suction from boilers.
4. Suction from main or reserve feed tanks.
5. Suction for hose connection.

The discharge manifold:

1. Discharge overboard.
2. Discharge to fire main.
3. Discharge to auxiliary feed system.
4. Discharge for hose connection.

There are extensions from the auxiliary feed main to the port boilers, and through the center-line bulkhead to the starboard boilers.

One of these pumps was in constant operation during the official trial and averaged about ten double strokes per minute, maintaining a pressure of about 425 pounds in auxiliary feed line. The pump discharged through its relief valve.

BLOWERS.

The fireroom blowers, manufactured by the B. F. Sturtevant Co., are located on the berth deck near the center-line bulkhead and discharge directly into the firerooms. A trunk extends from the blower room to the main deck for the supply of air.

In each blower room is located the air lock to the firerooms. To establish communication with the firerooms, a voice pipe is installed. The arrangement of the blowers is symmetrical

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with respect to the center-line bulkhead, and is as follows, beginning forward:

1 blower for single fireroom.
2 blowers for double firerooms.
2 blowers for double firerooms.
2 blowers for double firerooms.
1 blower for single fireroom.

Data for One Blower:

Diameter of fan, inches	60
Width of tip of blades, inches	18
Area of induction nozzle, square inches	1,280
eduction nozzle, square inches	1,580
Number of cylinders	2
Diameter of cylinder, inches	5
piston rod, inches	$1\frac{1}{8}$
Stroke, inches	4
Valve travel, inches	$2\frac{1}{4}$
Length of connecting rod between centers, inches	12
crosshead pin, inches	$2\frac{1}{4}$
Diameter of crosshead pin, inches	$1\frac{3}{4}$
Length of crank pin, inches	$3\frac{1}{2}$
Diameter of crank pin, inches	3
Area of crosshead slipper, square inches	8
Diameter of crank shaft, inches	$2\frac{15}{16}$
Length of bearings, inches	$6\frac{3}{8}$
Crank angle, degrees	180
Number of blowers on ship	16
blower rooms	10

Data from Official Trial:

Revolutions per minute	500
Air pressure in fireroom	1.17
I.H.P. for one blower	21.9
sixteen blowers	350

ASH HOISTS.

The arrangement of the fireroom ventilators corresponds to the arrangement of blower rooms. The ash-hoist engines are located in the air trunks for the blowers, and are bolted to the ventilator casing. Inside the ventilator shaft guides are placed for the ash buckets. There is an opening on the gun deck for discharging ashes, and in the forward ventilator shafts

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is an additional opening on the boat deck for supplying coal to the launches.

The engine is operated by a follow-up gear from the gun deck. On the test made the engines hoisted 300 pounds from the fireroom floor to the gun deck in five seconds with 80 pounds of steam. Under the main deck is placed a trolley track to facilitate transporting ashes to the ash chutes. On the port side are located three of the ash chutes and the other three are on the starboard side.

Data for One Engine:

Number of cylinders	2
Diameter of cylinder, inches	4½
of piston rod, inches	1
Stroke, inches	4½
Valve travel, inches	1⅛
Length of connecting rod between centers, inches	10
crosshead pin, inches	1½
Diameter of crosshead pin, inch	1
Length of crank pin, inches	1½
Diameter of crank pin, inches	1¾
Area of crosshead slipper, square inches	11
Diameter of crank shaft, inches	2
Length of bearing, inches	3½
Crank angle, degrees	90
Diameter of drum, inches	5
Length of drum, inches	10
Number of ash-hoist engines	10

EVAPORATING PLANT.

The evaporating plant is arranged with two evaporators, the pumps and fresh-water tank on the berth deck. On the gun deck, directly above, are the other two evaporators and filters. In the ventilating trunk the three distillers are placed vertically.

Evaporators.—The shells and heads are of plate steel. The heads are supported by stay bolts and I bars riveted to them. Brass tubes expanded into composition headers provide the heating surface. The passage of the steam is from the front to the back header through all the tubes except the bottom row, through which it is returned to the front header. Steam gauge, relief valve and drain connections are made to the front

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header. A trap, which discharges to the feed tank, is provided to drain each evaporator.

Distillers.—These are the Navy Department design. The bodies are of cast iron and the cooling surface is obtained from brass tubes, tinned inside and out, and expanded into brass tube sheets. Expansion is provided for by bolting a casting with a nozzle to the upper tube sheet. This nozzle goes through a stuffing box at top of distiller. The circulating water passes through the tubes and out at the nozzle.

Pumps.—The distiller circulating water is supplied by a 12-inch by 16-inch by 16-inch horizontal, simplex, piston pump. This pump may discharge to the sanitary system or to the fire main. The evaporator feed is supplied by a 4½-inch by 5-inch by 6-inch vertical simplex pump; its suction is from the discharge pipe for the distiller circulating water. The fresh and salt-water pump is of the vertical, simplex type and of the following dimensions: 6 inches by 5 inches by 4 inches by 6 inches. The cylinders are arranged with the steam at the top, below it the fresh water, and the salt water at the bottom. All pumps are of the Blake design.

Piping.—Each evaporator receives steam through a reducing valve on the auxiliary steam line. The vapor may be sent to the distillers, to the auxiliary exhaust or to operate the evaporators in "double effect." From the distillers the fresh water passes through the filters to the fresh-water tank, from which it is pumped to the distributing tank, or drains to the reserve feed tanks.

A test was made during the run from New York to Boston to determine the capacity of the plant. The specifications require that the evaporators shall have a combined capacity of 23,000 gallons of potable water per twenty-four hours, and the distillers to have a combined capacity of 10,000 gallons of potable water per twenty-four hours. Two evaporators were used for this test and all three distillers. The discharge nozzle of each distiller was tapped to permit testing the distilled water with the silver-nitrate solution. The filtering material was removed from the two filter tanks,

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and these were filled alternately with the distilled water. A record was made of the number of times these tanks were filled. The evaporator, feed, and fresh and salt-water pumps were in constant operation, maintaining an average hydrometer test of $\frac{3}{32}$ at a temperature of 190 degrees Fahrenheit. The following data was obtained:

EVAPORATORS.

Data for One Evaporator:

Heating surface (tubes only), square feet	290
(headers only), square feet	33
total, square feet	323
Gallons of water evaporated in 24 hours	7,500
Gallons of water evaporated per square foot of tube-heating surface in 24 hours	25.9
Gallons of water evaporated per square foot of total heating surface in 24 hours	23.2
Gallons of water evaporated per 24 hours for four evaporators	30,000

DISTILLERS.

Data for One Distiller:

Cooling surface (tubes only), square feet	66
(tube sheets only), square feet	2.2
total, square feet	68.2
Gallons of water distilled in 24 hours	5,000
Gallons of water distilled per square foot of cooling surface in 24 hours	73.3
Gallons of water distilled per 24 hours for three distillers	15,000

Data from Observations (Averages):

Temperature of injection water, degrees Fahrenheit	49
discharge water, degrees Fahrenheit	74
room, initial at 8 P. M., degrees Fahrenheit	88
final at 2 A. M., degrees Fahrenheit	63
Strokes of circulating pump per minute	36

REFRIGERATING PLANT.

The refrigerating plant is located on the port side of the berth deck forward. The refrigerator rooms are divided into five compartments, and are insulated with cork, lined on the inside with zinc, except the floors, which are covered with sheet lead. Above, on the gun deck, are located the two scuttle butts.

The ice machine is of the Allen dense-air type, designed for a cooling effect equivalent to three tons of ice per twenty-four

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hours. The piping for the cooling air is arranged so that it may be sent through the refrigerator rooms and returned to the machine, or it may be sent through the scuttle butts before returning to the machine. It may also be by-passed through the ice tank before it goes to the refrigerator rooms.

When the test for this plant was made the cooling air was sent through the ice tank, thence to the refrigerator rooms and to the scuttle butts. At the start the temperature of the water in the ice tank was 49 degrees Fahrenheit, and in two hours and twenty minutes six cakes of ice had been made. At this time the three cakes at the end, where the cooling air is exhausted, were not completely frozen, so their positions were exchanged with the three middle cakes, and were completely frozen thirty minutes later. The temperature of the water put into the scuttle butts was noted, and also at the end of thirty minutes when seventeen gallons were drawn from each. Afterward the scuttle butts were refilled, and this operation repeated every thirty minutes during the test. At the end of one hour the number of gallons drawn off was increased to twenty-three.

The following are the averages of the data obtained at the end of four hours:

Temperature of the water to be frozen, degrees Fahrenheit	49
Pounds of ice in 24 hours	1,080
Temperature of water put into scuttle butts, degrees Fahrenheit	56
drawn off scuttle butts, degrees Fahrenheit	39
Number of gallons drawn off in 24 hours	2,200
Temperature of room, degrees Fahrenheit	65
atmosphere, degrees Fahrenheit	46
injection water, degrees Fahrenheit	51
ICE MACHINE.	
Steam pressure, pounds	235
Compression cylinder, pounds	241
Expansion cylinder, pounds	64
Revolutions per minute	108
REFRIGERATOR ROOMS.	
Temperature entrance, initial, degrees Fahrenheit	54
final, degrees Fahrenheit	50
Junior officers' room, initial, degrees Fahrenheit	40
final, degrees Fahrenheit	1
Ward room, initial, degrees Fahrenheit	51

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Ward room, final, degrees Fahrenheit	10
Captain's room, initial, degrees Fahrenheit	48
final, degrees Fahrenheit	—2
Crew, initial, degrees Fahrenheit	48
final, degrees Fahrenheit	17
Ice Machine Data:	
Diameter, steam cylinder, inches	10
compression cylinder, inches	9
expansion cylinder, inches	7½
piston rods, inches	2
Stroke, inches	7
Length of connecting rod, center to center, inches	33

ELECTRIC PLANT.

The electric plant is located below the berth deck and forward of the firerooms. It extends the full width of the firerooms and is divided into an upper and lower dynamo room. The center-line bulkhead passes through each, and an archway through it provides communication between port and starboard sides. The General Electric marine generating sets are installed. In the lower dynamo room are located:

Starboard: 1 100-kw. marine generating set.
1 50-kw. marine generating set.
1 dynamo condenser.
1 exhaust separator.
1 steam trap for separator.
Port: 2 100-kw. marine generating sets.

In the upper dynamo room are located:

Starboard: 2 50-kw. marine generating sets.
1 switchboard.
1 steam separator.
Port: 1 50-kw. marine generating set.
1 steam separator.
1 switchboard for searchlights.
1 central communicating room.

Piping.—Steam is supplied on each side of the ship and passes through a separator and a reducing valve to a distributing manifold. There is a pipe cross connecting these mani-

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folds and another to the lower dynamo room. The engines exhaust to the dynamo condenser, or to the auxiliary exhaust to the engine room or the atmosphere. The separators drain to a trap which discharges to the main feed tank, and the cylinder drains discharge to the dynamo condenser or exhaust separator. The dynamo exhaust drain, for use when exhausting to the atmosphere, also discharges to the exhaust separator. A 2-inch by 1¼-inch by 2¾-inch pump is provided to keep this separator drained, and discharges to the main feed tank. The dynamo condenser air pump discharges to the main feed tank in the starboard engine room.

Data for Condenser:

B.W.G. of tubes	18
Number of tubes	1,001
Diameter of tubes, inch	⅝
Length of tubes, feet and inches	7- 8
Cooling surface, square feet	1,228
Horizontal combined air and circulating pump, inches	12 × 14 × 14 × 12

Data for Electric Sets:

		100 kw.	*50 kw.*
Diameter of	H.P. piston, inches	10	7½
	L.P. piston, inches	18	14
	piston rod, inches	2½	1¾
Stroke, inches		10	8

SYSTEM OF WIRING.

The cables from each generator are led to the switchboard and bus-bars connect this board to distributing panels. The distributing panels for light and power are separate. By this means the output of any generator may be sent to the light-distributing panel or to the power-distributing panel, or any number of machines may be so used. A special distributing panel is installed for the searchlights, and the output of any generator may be sent to it.

MACHINE SHOP.

The machine shop is located beneath the berth deck and between the engine and fire rooms. It extends the full width of the engine rooms and is entered from the first grating in each engine room. An archway in the center-line bulkhead

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provides communication between the port and starboard sides. The following machines are installed:

1 30-inch swing extension gap lathe.
1 14-inch swing screw-cutting lathe.
1 16-inch crank shaper.
1 16-inch drill press.
1 12-inch two-wheel emery grinder.
1 16-inch sensitive drill.
1 30-inch grindstone.
1 combined hand punch and shears.
1 universal milling machine.

These tools are provided with the requisite attachments. Lockers are provided, also tool benches, and for the indicators two lockers are provided, one port and one starboard. The machines are driven by a motor located on the athwartship bulkhead. A test was made to ascertain if this motor was sufficiently powerful for the work to be required of it. In each tool a piece of steel was placed and the machine run at its heaviest cut. The emery wheels and grindstone were given a load by grinding a piece of steel. These operations occurred simultaneously and were maintained for one half hour.

The following data was obtained:

Volts, average	126
Ampères, average	23
Revolutions per minute	608
Electrical horsepower, input	3.9

BUILDERS' TEST OF MOTOR.

Electrical horsepower, input	17.1
output	14.5
Ratio	.85

THE OFFICIAL TRIAL.

The U. S. S. *Pennsylvania* left her anchorage in Boston harbor at 6·50 A. M., November 23, 1904, for the Cape Ann trial course. On her way out the forced-draft blowers were started, the boiler pressure increased and the main engines gradually worked up to full power. The ship was steered directly for the course, and, by 75th meridian time,

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	Hr.	*Min*	*Sec.*
Passed the first stake boat, north, A. M.	8	52	11
last stake boat, north, A. M.	10	51	34
Time required for turn	0	11	57.5
Passed the first stake boat, south, A. M.	11	3	31.5
last stake boat, south, M	12	59	37.5
Length of trial course in nautical miles			44.019
Corrected speed for the run north, knots per hour			22.300
south, knots per hour			22.575
Mean corrected speed			22.436

After passing the last stake boat the engines were gradually slowed down to 120 revolutions per minute for maneuvering the ship. The time required to put the helm from hard-a-starboard to hard-a-port was 20.5 seconds, and from hard-a-port to hard-a-starboard was 18 seconds. During this test the maximum angle of heel observed was 4.5 degrees. When the engines had been slowed down to 110 revolutions per minute they were stopped and reversed to full speed astern—the forced-draft blowers had in the meantime been stopped—and the time elapsed from the stopping of the engines until the ship was still in the water was two minutes and two seconds.

The draught of the U. S. S. *Pennsylvania*, taken while at anchor on the morning of the trial, was:

Forward, feet and inches,	24– 1.5
Aft, feet and inches,	24– 5.8
Mean, feet and inches,	24– 3.6
After the run:	
Forward, feet and inches,	23– 8.9
Aft, feet and inches,	24– 0.5
Mean, feet and inches,	23–10.7
corrected, feet and inches,	23–11.8

THE ENGINE ROOM.

The main engines during the trial ran with very little vibration and none of the bearings heated. The water service was used in the main bearings, crosshead guides and thrust bearings, but very little was used on the crank pin or crosshead bearings. The main circulating and air pumps worked

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smoothly. The maximum pressure permitted in the H.P. valve chest was 250 pounds by gauge, and during the entire trial the throttle valve was never more than .30 open.

Data from Trial:	*Port.*	*Starboard.*
Steam pressure at throttle, gauge	278.5	280.
in fireroom, gauge	300.2	300.2
H.P. valve chest, gauge	250.	250.3
M.P. receiver, gauge	107.5	114.4
L.P. receiver, gauge	23.3	23.9
Vacuum, inches	26	26.7
Position of links	Full gear.	
Temperature of atmosphere, degrees Fahrenheit	59.5	59.5
at floor plates	70.1	68.6
first grating	83.2	82.8
Total number of revolutions on run north	15,230	15,113
south	14,970	15,053
while on course	30,200	30,116
Average revolutions per minute, north	127.98	127.00
south	129.07	129.77
while on course	128.53	128.38
Indicated horsepower of main engines	14,336.4	13,669.7
total	28,006.1	

THE FIREROOMS.

It was desired for this trial that the coal consumption be accurately determined, and for this purpose the coal had been put in bags of 100 pounds each. The coal used was hand-picked Pocahontas. Permission was granted to set the safety valves at 315 pounds, but the pressure in the boilers was to average 300 pounds, the designed working pressure. This arrangement was to save feed water, because on her sister ship, the U. S. S. *Colorado*, the safety valves—set at 300 pounds—were lifting continuously throughout the trial, and this would materially affect the coal-consumption figures. In order to facilitate firing the furnaces they were numbered from left to right: 1, 3, 5, 2, 4 and 6. This would make number 1 furnace on opposite corners in the double firerooms and give the men ample room in which to work. An apprentice boy was provided with a watch, and instructed at what time intervals the furnaces were to be fired. At the termination of each interval the boy called the number of the furnace to be fired—

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these numbers were called consecutively. By this method the boilers were fired alternately. The time interval was one, two and three minutes, varying with the steam pressure, and in one of the firerooms the interval was six and even twelve minutes.

The forced-draft blowers were maintained at 500 revolutions per minute, except at the beginning of the trial when they slowed down to 300 for a short time. But it was decided to regulate the draft by the ash-pit doors, and the blowers were run at 500 revolutions per minute for the remainder of the trial. In one of the firerooms the ash-pit doors were closed the greater part of the trial, and closed at intervals in all of them. The thickness of the fires was about 6 to 8 inches and the coal was spread evenly over the fires.

The number of men in the fireroom was as follows:

Double firerooms:	Single firerooms:
1 Water tender.	1 Water tender.
4 Firemen.	2 Firemen.
2 Coal passers in fireroom.	1 Coal passer in fireroom.
2 Coal passers in bunker.	2 Coal passers in bunker.
1 Apprentice boy.	1 Apprentice boy.

The automatic feed-regulating valves worked almost without attention and in some firerooms were not touched during the trial. During the trial photographs were taken every 10 minutes of the smoke emitted from the smoke pipes. The color of the smoke was a dull brown to gray, and not unusually heavy. The temperature at base of smoke pipe did not at any time exceed 640 degrees Fahrenheit.

Data from Official Trial:

Forced-draft blower R.P.M., average	500
Air pressure, inches of water, average	1.17
Draft at base of smoke pipe, inches of water, average	—.34
Temperature at base of smoke pipe, average, degrees Fahrenheit	630
Thickness of fire, inches, average	6 to 8
Steam pressure, gauge, average	300.2
in port engine room, gauge, average	278.5
starboard engine room, gauge, average	280.0
port H.P. valve chest, average	250.0
starboard H.P. valve chest, average	250.3

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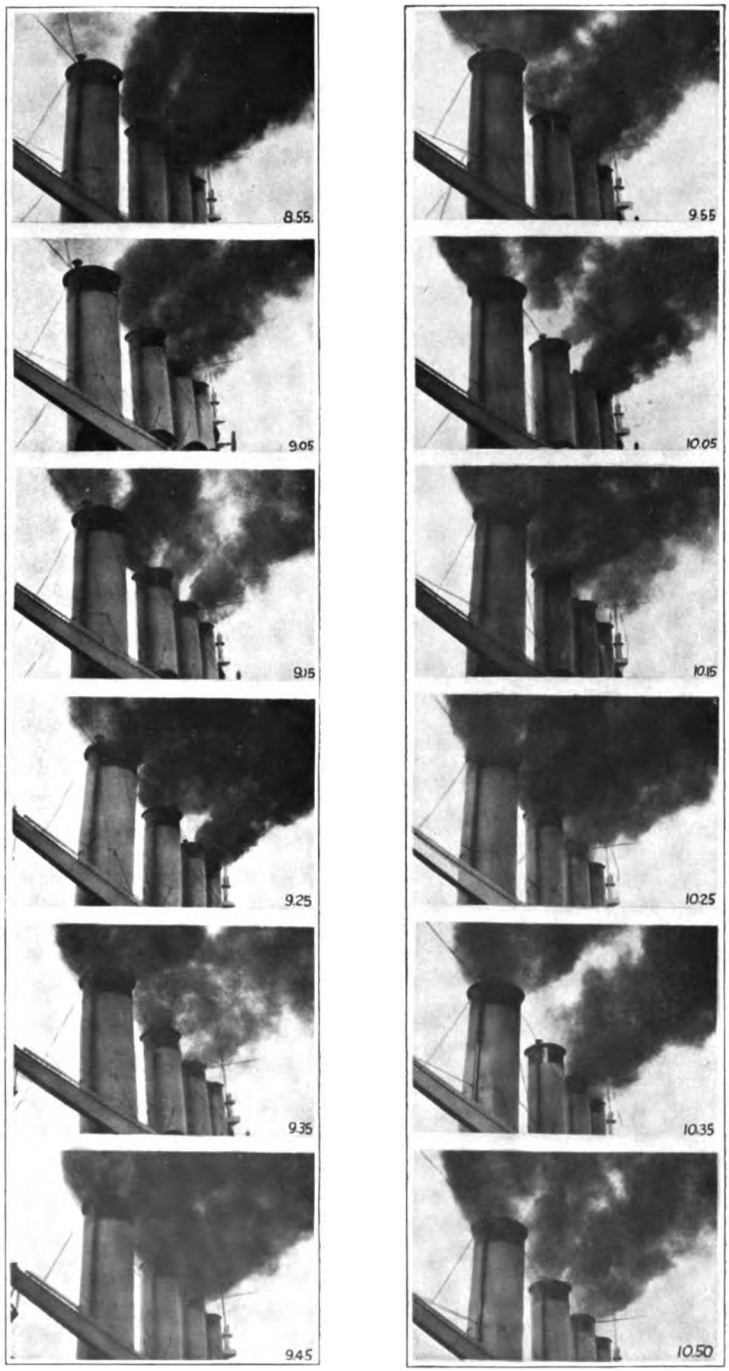

U. S. S. "PENNSYLVANIA."

Photographs of Smoke taken at ten-minute intervals during trial. Run North (Cape Ann towards Cape Porpoise).

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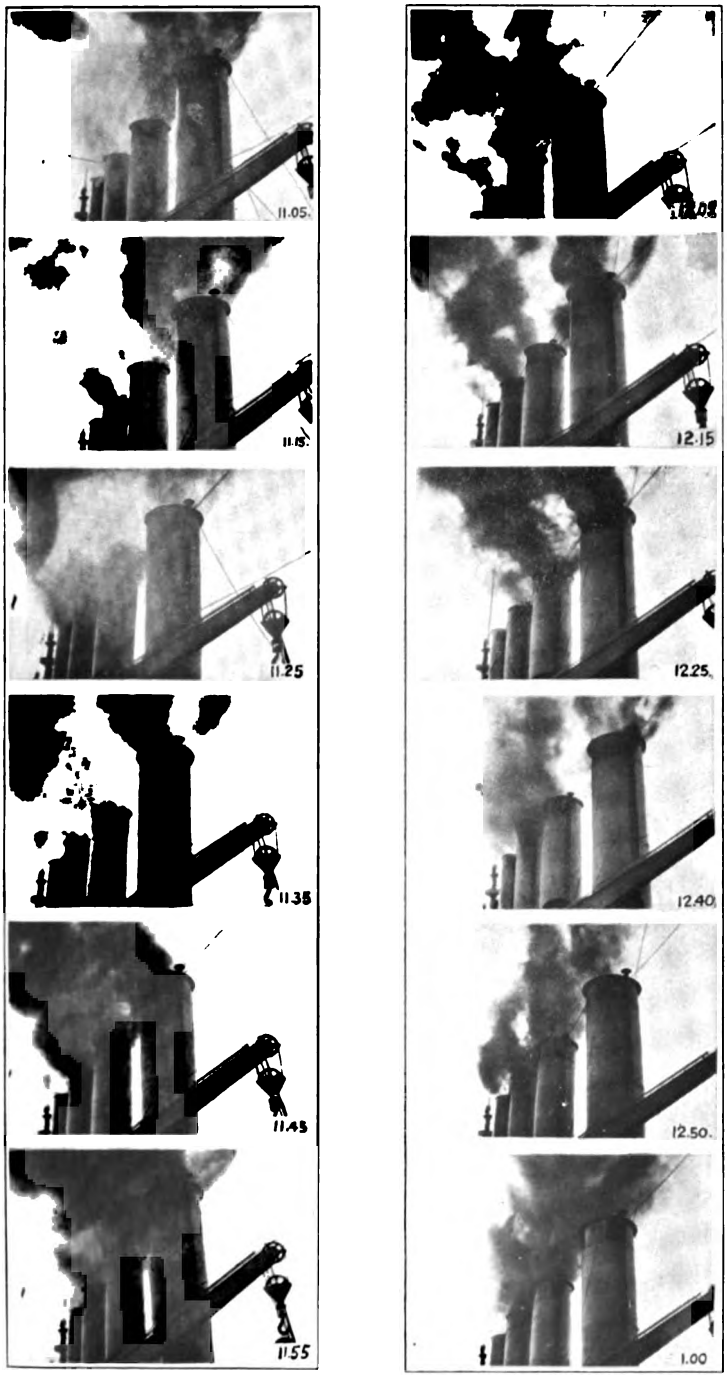

U. S. S. "PENNSYLVANIA."

Photographs of Smoke taken at ten-minute intervals during trial. Run South (Cape Porpoise towards Cape Ann).

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INSPECTION AFTER TRIAL.

Main engine cylinders, valve chests, low-pressure valves, crank and crosshead-pin bearings, crank-shaft bearings and crosshead guides were examined and found to be in excellent condition. The condensers and feed-water heaters were found to be tight and free from grease. The main air pumps, feed tanks, auxiliary condensers and pumps were free from grease and in good condition. The boilers, grates and brickwork were found to be in excellent condition. Some of the tubes in the lower rows were found to be bowed up from $\frac{1}{16}$-inch to $\frac{5}{16}$-inch, but generally were found to be in a normal condition. Several tubes were removed and no difficulty was encountered in removing them. The time required to remove a circulating tube was 58 seconds and a generating tube 1 minute and 3 seconds, or a total of 2 minutes and 1 second. The tubes were found to be in excellent condition.

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NEW ENGINEERING AND NAVAL CONSTRUCTION BUILDING, U. S. NAVAL ACADEMY.

By JOHN K. BARTON, COMMANDER, U. S. N.,

Head of Department of Engineering and Naval Construction, Naval Academy.

While the changes in the course of instruction at the Naval Academy, made necessary by the Personnel Law of 1899, have been watched with eager interest by the professional world both at home and abroad, none the less has been the interest displayed over the building operations in the transition from the old to the new and modern Academy, a work made necessary in order to accommodate the great increase in the number of midshipmen, together with the installation of apparatus, tools and machinery for their education.

By most of the members of the American Society of Naval Engineers and the readers of the JOURNAL, the removal of the old Steam Building of 1866, a tinge of regret may be felt at the disappearance of an old land mark, yet the transition to the commodious and roomy structure of the Engineering and Naval Construction building of 1904, with its facilities, roomy space and modern appliances, is the culmination of a hope that has been looked forward to for many years.

The cramped and restricted area of the old Steam Building has been felt severely, even with the limited number of students of a few years ago, averaging about 300, a number hardly exceeding the fourth class of today, while the entire number now under instruction in all classes is 825. With the entry of the class in June next the entire number of midshipmen will amount to nearly a thousand, which will probably be the maximum number in any one year.

With this large increase in prospect the design of the engineering building was executed, and as constructed has a

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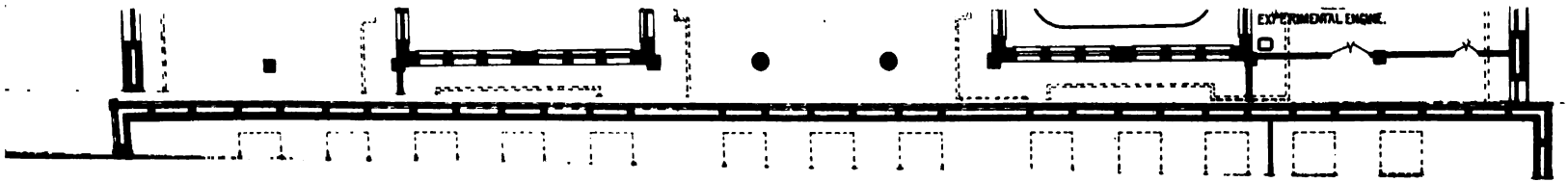

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capacity to accommodate all midshipmen in their studies of engineering and naval construction from their entrance into the Academy until their graduation.

In preparation for entrance into the new building a large addition has been made during the past year in the shape of electrically-driven tools, apparatus and machinery, especially in the model room, machine shop, laboratories and wood-working shop; while for the succeeding years the proposed plan is for a gradual installation of new material yearly from an appropriation deemed suited to the needs, until the building is thoroughly equipped.

For the model room numerous additions have been made in the various appliances in use for marine engines and auxiliaries, and, through the generosity of various engineering companies, material, and sectional and working models have been kindly donated. In addition, all the standard pumps in use in the Navy have been installed.

In the wood-working shop new lathes and other tools have been installed, bringing the shop to the present working capacity of fifty midshipmen.

In the drawing room the outfit has been enlarged by models on a reduced scale of various parts of actual engines used in the United States Navy, but with the number of midshipmen under instruction at the same time, the equipment in this respect is still far below the needs. With the progressive plan of installation, however, the defect in material of instruction in this, as in other branches of the engineering department, it is hoped will be remedied at an early date.

In the following description, together with the accompanying plates, the general arrangement, the features of the building and its capacity may be seen.

The main building is located in a northwesterly direction, on the axis of the proposed academic building and adjacent to the power-house pier. It is 266⅔ feet long, 175 feet wide, and is two stories in height, except the central part, between the interior courts and southeast fronts, which is three stories, the third story containing the mold loft. The roof is cov-

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ered with slate and has hinged skylights on the exterior and court sides; there is also a large ridge skylight over the mold loft.

In the course of construction and not yet completed is a wing or annex on the northwest side, 116 feet long, extending 66 feet to the rear, and in communication with the machine shop. In this wing will be located the boiler room, foundry, forge shop and copper shop, with the proper equipment of a modern plant.

Main Building.—The total floor space in the main building is approximately 91,450 square feet, distributed as follows:

First floor, square feet, ...	39,100
Second floor, square feet, ...	39,100
Third floor, square feet, ...	13,250
Total, square feet, ...	91,450

The principal entrance is on the southeast side and faces an extension of College Avenue of the City of Annapolis. Additional main doorways for freight and material are located in the remaining sides of the building and communicate directly with the the shops; the main entrance leads through the model room to a large divided staircase leading to the upper floors.

The construction of the building is of ferro-concrete and thoroughly fireproof throughout; the exterior walls are of light gray brick with granite trimmings. Two light wells, 75 by 50 feet, open at the top and terminating in courts on the first floor, contribute to make the building well lighted throughout its interior.

First Floor.—The main entrance leads directly into the spacious model room, the shape of the latter being that of an inverted "T" and taking in two sides of each light well previously noted. This room contains types of the different auxiliaries, air, circulating, feed pumps, etc., found on board naval vessels, sectional models, valves and valve gearing, models of triple and quadruple-expansion engines, indicators,

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types of steam-turbine vanes, model of steam-steering and capstan engine, full-size working model of the U. S. S. *Connecticut's* reversing engines, reversible ash-hoist engines, sectional and working models of steam-launch engines, working models of valve gear and link motion, elements of Niclausse, Babcock & Wilcox and Cahall boilers, together with the various naval construction models in the way of sections of various types of ships, working models of machinery under the cognizance of the Bureau of Construction and Repair. Various instruments of precision, fuel and gas-testing apparatus are contained in the cases surrounding the sides of the room.

Adjacent to the model room on the northeast side is the mechanical laboratory, divided into two parts; total area 125 feet by 50 feet.

The present equipment of the laboratory consists of:

1 Rand air compressor 10 inches by 14 inches, duplex steam, 16 inches by 10 inches by 14 inches compound air.

2 Wheeler condensers.

1 Triple-expansion experimental engine, with tank.

1 Dense-air ice machine, 1 ton.

1 Ammonia ice machine, 2 tons.

1 Evaporating plant, with pumps and distiller.

1 16-kw. marine direct-connected generator.

1 25-kw. Curtis steam turbine, with generator.

1 15-H.P. De Laval steam turbine.

1 Riehle testing machine.

1 Complete set of pneumatic tools, with all attachments.

2 Engines for disconnecting, making adjustments, etc.

1 Root blower.

Prony brakes, dynamometers, and engine and boiler-testing appliances.

Adjacent to the opposite side of the model room is the woodworking shop, 108 feet by 50 feet. The present equipment of this shop consists of 18 woodworking lathes, 1 power-driven band saw, 1 power saw bench, several hand-power tools, and benches and equipment for 32 midshipmen at bench work, making a capacity at present for 50 midshipmen, or, with

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the installation of additional tools, a capacity of 75. The machinery is driven by a Crocker-Wheeler motor.

Behind the model room, woodworking shop and laboratory, and extending the entire length of the building, is the machine shop, 266⅔ feet long by 50 feet wide, fitted with store rooms, lavatories and tool rooms.

Equipment.—All machine tools, lathes, etc., in the old Steam Building found to be in good condition were transferred to the new building, arranged in two groups, belted up and driven by two separate group motors. All new installations were made with individual motor drives, and as far as possible of the different standard makers of both machines and motors, in order to give the midshipmen the opportunity of observing the different mechanical movements and to assist in the class-room instruction.

The equipment consists of:

37 Lathes from 14-inch swing up to 27-inch swing.
4 Planing machines.
6 Shaping machines.
4 Drill presses.
3 Sensitive drilling machines.
3 Milling machines.
1 Combined drilling and slotting machine.
3 Slotting machines.
1 Radial drill.
1 Spliner.
1 Grinding machine.
2 Boring machines.
1 Pipe-cutting machine.
3 Tool grinders.
1 Milling cutter and tool grinder.
1 Drill grinder.

Of these 15 lathes, 2 shaping machines, 1 drill press, 3 sensitive drills, 2 milling machines, 2 slotting machines, 1 radial drill, 1 planing machine, 1 combined slotting and drilling machine, 2 tool grinders and 1 drill grinder are motor driven.

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To meet the large number of midshipmen in the lower classes as they reach the first and second classes an increase in machine tools and lathes will be required in the coming year, as with the present number all machines are constantly occupied.

At the end of the shop a space is reserved for a small erecting floor when its services will be required, while at the opposite end are placed tool rooms, extending the width of the shop, and separated from it by expanded metal screens.

A vise bench lines one length and width of the shop, and is fitted with 72 vises for the use of midshipmen engaged in bench work, the present complement now being 70 midshipmen, at this work.

The total complement at present engaged in the afternoon exercises in the machine shop, wood-working shop and laboratories is 193. In the coppershop, foundry, forge and boilershop in the old Steam Building there are at the same time 57 midshipmen, together with 30 engaged afloat in handling the machinery of the monitors *Nevada* and *Arkansas*, or the two torpedo-boat destroyers *Lawrence* and *Truxtun*, making a total of 280 midshipmen engaged in practical engineering instruction each day.

Surrounding the first floor is a large pipe trench containing all the steam, water and air piping connected to the various machines where required. From the air receiver of the air compressing machine a discharge pipe leads completely around the machine shop, laboratories and model room, with taps placed at convenient intervals, by means of which hose connections can be made to pneumatic tools in any of these shops, or connections made quickly and readily to models and machines desired to be operated for instruction purposes.

Second Floor.—In the center of the southeast side is the office of the head of department, and on either side, connected by communicating doors, are the instructors' rooms and a reference library and reading room. On this floor are nine commodious and well-lighted section or class rooms, two store rooms, two photographic dark rooms, and a toilet and coat room.

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The drawing room is also situated on this floor, and consists of the entire rear and northeast wing, giving a floor area of 19,583 square feet, and, with drawing tables spaced 7 feet apart from center to center, accommodates a class of 325, leaving ample room for model tables, drawing-board cases, sinks, blue-print rooms and passage ways. This room can be still farther increased by throwing into communication with it, by means of rolling wood partitions, the southwest wing, now used as an examination room, 125 feet by 50 feet, and giving accommodation for 105 more drawing tables or for a class spaced for examination alone of 300 midshipmen. The actual number of midshipmen at present in these two rooms for examination in mathematics and engaged in mechanical drawing is 575.

The drawing room is well lighted by means of overhead skylights, outboard windows and windows opening into the light wells. For late afternoon work or winter-weather conditions portable electric lights lead from sockets under each drawing table. In addition to the ordinary blue-printing outfit of tracks and frames for sunlight printing a large electric blue-printing machine is installed, by means of which each midshipman can do his printing irrespective of weather conditions, and with a great saving of time, which is rather an important consideration in the course of mechanical drawing. Midshipmen are given free access to the drawing room, whenever their other duties permit, for any increased work in mechanical drawing they may desire.

Third Floor.—This floor has been designed as a mold loft, for the laying off of ships, the preparation of molds, templets and the usual work done in a mold loft. It is also fitted with draughtsman's tables for the use of upper classes in designing work and in illustrating the practical problems given out in the section rooms.

Electrical Work.—The building is very thoroughly equipped electrically. The lights are ample and sufficient, with convenient panels and switches, and include a complete bell and telephone system, together with the wiring necessary

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for power purposes. Enameled iron-pipe conduits are used throughout for all wires of the light, power and call-bell services. The wiring is on the three-wire system throughout, except branch lamp circuits, which are of two wires.

The power circuit is divided into six circuits, consisting of one set of mains each for the following: The machine shop for the machinery motors, the woodworking shop, the model room, one to each experimental laboratory, and one for future use in a freight lift in the machine shop. These mains are large enough to supply current for all the machinery motors in use at present and for the future installation.

From the office of the head of the department interior telephones lead to all the shops, class rooms, and instructors' rooms.

Shop Building.—The shop building or annex to the Engineering and Naval Construction building is located in the rear of the main building on the axis of the same.

The building is one story in height, with a covered terrace connecting it with the main building, and the general floor is directly on the ground surface.

The structural parts of the building, including interior and exterior columns, foundation piles, wall supports, cupola, charging platform, and the entire building with the exception of the lantern, is of ferro-concrete construction and fire proof throughout.

When completed the building is to contain the foundry, forge and boiler shops, cupola, platform and lift to the same, brass furnaces, pipe trenches, etc. The floors are to be of concrete and cement, excepting that portion of the foundry requiring a dirt floor.

Forge.—Referring to the plans of the annex, the forge shop has an area of 50 feet by 65 feet with a down-draft forge equipment of twenty-eight forges, with the necessary outfit of exhausters, blowers, anvils, forge tools, vise benches, etc., a steam hammer with attachments, and an oil-annealing and hardening furnace.

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The total capacity of this shop for instruction will be about forty midshipmen.

Foundry.—The foundry is 66 feet by 32 feet, and the present designed equipment is for a small cupola complete with blower, elevator to charging door, core oven, cranes, crane ladles, and a complete outfit of molding implements.

In the shop will also be installed brass furnaces for use in brass casting, and consisting of one or two Rockwell rotary melting furnaces complete. Oil fuel will be used in this connection with oil burners, compressor or pressure blower, storage tank, the necessary piping, oil pump and ladle heater.

One section of this shop will be devoted to elementary instruction in coppersmithing in the ordinary work usually found on board of a naval vessel. Capacity, about forty midshipmen.

In the rear of the foundry is the boiler shop, 66 feet by 32 feet, to be equipped with a Ward round boiler, and two proposed small units of the typical water-tube boilers found on board battleships and armored cruisers of our Navy. As the drive of all the machine tools in the department is electrical, these boilers will be of small capacity, about 75 or 100 H.P. each (commercial), to be used for boiler-test instruction, and for driving the various auxiliaries, ice machines, air compressors, experimental engines, etc., in the laboratories.

In addition to the above installation the shop will be fitted as a boiler-repair shop for the instruction of midshipmen, in which the various operations of shell boiler construction and repair may be illustrated, together with the work of construction and repair of water-tube boilers. Space and material can be provided for the instruction of about forty midshipmen.

Cost of the Structure.—The cost of the engineering building and annex, including foundation and preparations for the same, but without furniture and equipment, is as follows:

For the main building, . .	$360,000.00
annex, . . .	40,000.00
Total, ...	$400,000.00

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With the completion of the annex, its installation of tools and the continuance of the work of fitting up the shops, laboratories and model rooms of the main building, all of which will probably be well finished during the coming year, and with the present system of summer practice cruises where the midshipmen cruise in a squadron composed of types of ships from a destroyer to a battleship, with engineering duty on each type, the facilities offered for practical engineering training have never been equaled by any class previous to the enactment of the Personnel Law, whether of the engineers or of the line.

With this elementary training at the Naval Academy, the suggestion outlined in the annual report of the Chief of Bureau of Steam Engineering, 1904, as to the necessity of giving a sufficient quantity of engineering duty on board ships to all line officers, at first in a subordinate capacity, to render them qualified to take charge of the engineering departments of any and all ships, and to select annually from among the list of ensigns a certain number of officers for the performance of engineering duty alone, would seem to offer a practical solution to the problem of obtaining an efficient engineering personnel for duty on our battleships and cruisers, furnish the professional training for the inspection of structural work and for the management of the engineering department in our navy yards and shore stations.

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CONTRACT TRIALS OF SHIPS AS EXPERIMENTS UPON PROPELLERS.

By John Lowe, Rear Admiral, U. S. N., Retired.

Anyone interested in the sea and in ships can find in the Journal of the American Society of Naval Engineers much information of great value; but of all the information therein contained, that which is of most value to the naval architect and engineer is that which relates to the contract trials of naval vessels. Especially to that class engaged in the study of the screw propeller the information there furnished is of indispensable value, and to this class this paper is addressed, with the purpose of showing that this data can furnish complete experiments upon the screw propeller, unsurpassed even by tank experiments.

The method is a trial-and-error process, somewhat akin to the Sumner method in navigation, whereby the efficiency of the screw, its friction, the thrust, and so forth, can all be ascertained, beside simultaneously testing the method and means employed.

Let (*IHP*) represent the power applied to one screw to propel the ship. Of this power a portion is absorbed in engine friction, the amount of which, in the light of many experiments, lies in the neighborhood of 28 per cent., leaving 0.72 (*IHP*) as the net power to revolve the screw.

This net power by the elder Froude was termed the "turning moment," and we will accept this value 0.72 (*IHP*) for this moment and represent it by (Z).

Now (Z) is to be again subdivided into three components: (F) is the power consumed in friction and other resistance to the passage of the screw through the water.

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(S) is the power lost in slip.

(A) is the remainder, usefully applied to propulsion.

(δ) is the slip unit given in every trial.

Since $e = \frac{A}{Z}$ is the efficiency of the screw, we have $A = eZ$; $S = \delta Z$ and $F = (1 - (e + \delta))\, Z$, or, for brevity, $F = cZ$.

The problem is to find e, and therefore each of the subdivisions. The process will also test the method as well as the means employed.

The method requires us to systematically pair two ships against each other and compare the data obtained from their contract trials.

We select two pairs: the armored cruiser *Colorado*, of 14,500 tons, against the torpedo boat *Rodgers*, of 147 tons, as one pair, while the battleship *Ohio* will be paired with the torpedo boat the *Winslow*; the dissimilarity could hardly be greater.

For our convenience, we extract from the trials the data that we need, and tabulate as follows.

Symbol employed.	Z	R	δ	P	r	Φ
				Propeller.		
Vessel.	Turning moment.	Revolutions per minute.	Slip unit.	Pitch, feet.	Radius, feet.	Projected area, square feet.
Colorado	9,410	126.79	0.192	22	9	74
Rodgers	830.16	395.8	0.2094	7.82	2.625	7.93
Ohio	5,740	123.85	0.156	17.25	8.125	75
Winslow	830.16	393.92	0.1816	7.80	2.625	7.93
Kentucky	4,350	114	0.1293	17.25	8.375	69.2

The trial and error method now follows. First, we assume a larger value for (e) than it can possibly be, and tabulate it in column e of the following tables. In column δ we tabulate the unit slip obtained from the data, and in column (c) we tabulate $c = (1 - (\delta + e))$ for use with $F = cZ$, as previously explained. Next, we assume smaller and smaller values of

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For the *Colorado*.

e	δ	c	f	C
Efficiency assumed value.	Obtained from the trial.	$c = 1 - (\delta + e)$.	Logarithm.	Logarithm.
0.73	0.1920	0.078	—11.8023	13.0644
0.721	0.1920	0.087	—11.8487	
0.72	0.1920	0.088	—11.8537	
0.70	0.1920	0.108	—11.9426	
0.69	...	...	...	

For the *Rodgers*.

e	δ	c	f	C
Efficiency assumed value.	Obtained from the trial.	$c = 1 - (\delta + e)$.	Logarithm.	Logarithm.
0.66	0.2094	0.1306	—11.7887	12.2463
0.64	0.2094	0.1506	—11.8506	
0.6392	0.2094	0.1514	—11.8531	
0.63915	0.2094	0.15145	—11.8536	
0.6391	0.2094	0.1515	—11.8541	

For the *Ohio*.

e	δ	c	f	C
0.708	0.156	0.136	—11.8529	13.0394
0.709	...	0.1361	—11.8532	
0.7078	...	0.1362	—11.8535	
0.7077	...	0.1363	—11.8538	
0.707	...	0.137	—11.8561	

For the *Winslow*.

e	δ	c	f	C
0.69	0.1816	0.1284	—11.8257	12.2027
0.683	0.1816	0.1354	—11.8487	
0.682	0.1816	0.1364	—11.8529	
0.6815	0.1816	0.1369	—11.8539	
0.68	0.1816	0.1384	—11.8583	

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(e) until a smaller value is reached than can be expected, and fill the table as just explained.

Now, then, we employ the friction formula of the appendix, but right here we remark that this formula has not yet been accepted—it is yet upon trial along with the method; we accept it, however, provisionally.

Into this formula we insert the data obtained from the trial, and so obtain the value of (fC), as explained.

We then have the equation $cZ = fC$ to find (f), which we tabulate appropriately, and the tables are ready, and the problem is solved as follows.

We have assumed for (e) values which are too great, and have filled the tables with values smaller and smaller until the value is too small.

(1) Somewhere between these extremes lies the true value.

(2) There is only one true value for (f), all the others are in error.

(3) The correct value lies in both tables, that value which is the same in each is manifestly the correct value. In other words, each table being plotted as a line, each line intersects at the point where the value of $\log(f) = -11.8536$, giving for the *Colorado*, $e = 0.72$; for the *Rodgers*, $e = 0.639$.

Now, since these values accord with experience, and because of the coincidence of (f), we are justified in saying that the truth of the method of the friction formula and of the results are probably proven. When, however, we find the same coincidence in a second pair, as in the table for the *Ohio* and the *Winslow*, giving the precisely same value for (f), then we are justified in saying that probability ceases and certainty commences; and when many other pairs of which these two are only samples, give like results, then we are completely justified in asserting the methods, the means and the results, as demonstrated.

We have, therefore

$$e = 1 - \left(\delta + \frac{F}{Z}\right)$$

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in which

$$F = f\psi R^3 \left(\frac{P^4}{\pi\gamma} + \frac{8}{3}P^2\pi r + \frac{16}{5}\pi^3 r^3\right) : \log(f) = -11.8536$$

δ and Z taken from any trial under investigation.

Therefore, any single trial without reference to others may give a complete analysis of the performance of a screw propeller.

Example, the *Kentucky*, $e = 1 - \left(0.1293 + \frac{615}{4350}\right)$, $e = 0.7292$.

APPENDIX.

FRICTION OF A SCREW PROPELLER.

Imagine a metallic plane with the surface of (S) square feet, the thickness very small, immersed in water, and forced to move therein in the direction of its length.

It will experience a resistance measured in pounds equal to the effort required to maintain motion.

It is generally conceded that the law and the amount of this friction is expressed in the equation $F = fv^2S$, and the work done in overcoming this resistance is $Fv = fv^3S$ foot pounds per second.

The screw propeller consists of a large number of such planes, and the friction of the entire propeller is the sum of the friction of all these elementary planes.

Consider the work done upon one of these planes, situated in radius (r) feet from the center, in a propeller of (P) feet pitch revolving at the rate of $\left(\frac{R}{60}\right)$ revolutions per second, the length of the plane being (l) feet and its thickness dr feet.

The path of this plane will be in a helix, its velocity $v = \frac{R}{60}\left(P^2 + (2\pi r)^2\right)^{\frac{1}{2}}$ feet per second, and its surface $S = 2ldr$.

The work done per second upon this plane will be

$$dW = f\frac{R^3}{60^3}\left(P^2 + (2\pi r)^2\right)^{\frac{3}{2}} 2ldr, \quad . \quad . \quad . \quad . \quad (1)$$

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This is a complete algebraic statement of the method employed by Isherwood for measuring the friction of one element.

A large number of such elements being measured and figured were set up as ordinates, and the sum or integral of them all was obtained by any of many well known methods.

This method, though unsurpassed for accuracy, is exceedingly laborious in practice, even to a prohibitive degree.

This summation or integration may be more rapidly and more accurately obtained by means of the Integral Calculus.

Let (l) be some fraction (k) of the helix; then

$$l = k(P^2 + (2\pi r)^2)^{\frac{1}{2}},$$

which being substituted in equation (1) becomes

$$dW = 2f\frac{R^3}{60^3}k\left(P^2 + (2\pi r)^2\right)^{\frac{1}{2}}dr,$$

which being expanded becomes

$$dW = 2f\frac{R^3}{60^3}k(P^4dr + (8P^2\pi^2r^2dr) + 16\pi^4r^4dr.$$

Integrated,

$$W = 2f\frac{R^3}{60^3}k\left(P^4r + \frac{8}{3}P^2\pi^2r^3 + \frac{16}{5}\pi^4r^5 + 0\right)$$

foot pounds per second.

Now, in this equation, k occurs as a coefficient in single ratio, and since its value varies along the blade, it is necessary to obtain its mean value, which is easily proven to be $k = \frac{\Phi}{\pi r^2}$ where φ is the projected area of the screw.

Substituting and reducing to horsepower units, we get

$$F = \frac{2f\Phi R^3}{60^2 \times 33000}\left[\frac{P^4}{\pi r} + \frac{8}{3}P^2\pi r + \frac{16}{5}\pi^3r^3\right]$$

in horsepower units.

We may, however, collect all the constants into one, and write

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$$F = f\Phi R^3\left(\frac{P^4}{\pi r} + \frac{8}{3}P^2\pi r + \frac{16}{5}\pi^3 r^3\right),$$

and we can get the value of f as in the main paper,

$$\log f = -11.8536.$$

If, whenever speaking of the formula we desire brevity, it may be done by representing the bracket by B, and we write

$$F = f\Phi R^3(B).$$

If we desire to discuss and compare two propellers or more as to friction, then we obtain all these values, and write $F = fC$, which is the form used in the main paper.

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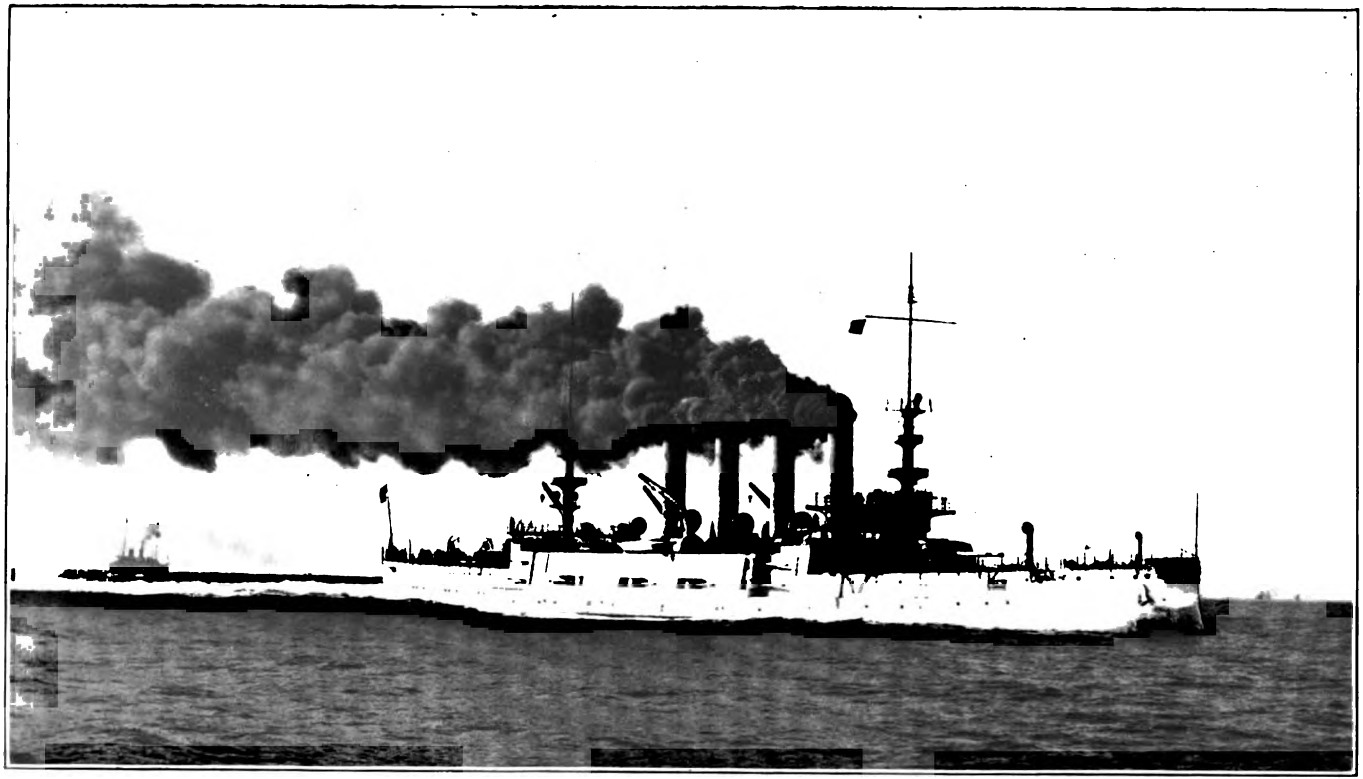

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U. S. Armored Cruiser "West Virginia."

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UNITED STATES ARMORED CRUISERS *WEST VIRGINIA* AND *MARYLAND.*

DESCRIPTION OF VESSELS.—OFFICIAL TRIAL PERFORMANCE.

By Lieutenant Commander W. Strother Smith, U. S. N., Member.

In accordance with an act of Congress appropriating for the increase of the Navy, the contract was signed on January 24, 1901, by the Newport News Shipbuilding and Drydock Company, for the construction of the hull and the machinery of the U. S. armored cruisers *West Virginia* and *Maryland.* As originally designed, the *West Virginia* was to be sheathed and her sister ship unsheathed, and the contract prices were $3,885,000 and $3,775,000 respectively. Subsequent to the signing of the contract it was decided between the contracting parties not to sheath the *West Virginia*, and there was made a decrease in the contract price of $135,931 for this vessel. The armor and armor bolts (exclusive of protected deck), ordnance and ordnance outfit and certain articles of equipage are supplied by the Government and their cost is not included in the contract price, but the expense of fitting and installing these is borne by the builders.

The date of completion by contract was fixed at January 24, 1904, but the time has been extended for just cause until February 6, 1905.

The speed called for was to be 22 knots per hour with a mean draught of 24 feet 6 inches and with an air pressure of 1 inch in the ash pits. Previous to the trial, the draft was changed to 24 feet 1 inch, and 2 inches of air pressure was allowed. It was stipulated that the vessels should be accepted if the speed did not fall below 20 knots at a decreased cost of $50,000 per quarter knot between 22 and 21 knots, and $100,000 per quarter knot between 21 and 20 knots per hour. Below 20 knots it was optional with the Secretary of the Navy to reject the vessel so failing or accept her at a cost to

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be agreed upon between the contracting parties. No bonus was to be allowed for any increase over the 22 knots.

The full complement of officers and men will be 808.

HULL DATA.

HULL.

Length between perpendiculars, feet	502
on L.W.L., feet	502
over all, feet and inches	503-10¼
Beam, extreme, feet and inches	69- 6½
at L.W.L., feet and inches	69- 6½
Ratio of length to beam	7.24
Depth (molded), feet and inches, main deck	41- 2
upper deck	48- 8¼
Height, flying bridge above main deck, feet and inches	24- 0
Draught, normal, mean, feet and inches	24- 1
Displacement, full load draught (empty bunkers), tons	13,226.4
normal draught (24 feet 1 inch), tons	13,720
full load draught (full bunkers), tons	15,170
per inch at L.W.L., tons	58.15
Area midship section to L.W.L., square feet	1,598
L.W.L., plane ( ), square feet	24,380
Wetted surface, square feet	42,850
Center of gravity of L.W.L. plane, aft of M.S., feet	7.85
buoyancy above bottom of keel, feet and inches	13-4¼
aft of M.S., feet	3.13
gravity above bottom of keel, feet	——
Transverse metacenter above center of buoyancy, feet	14.94
Longitudinal metacenter above center of buoyancy, feet	652.5
Coefficient of fineness block	.569
of M.S	.954
L.W.L	.696
Cylindrical coefficient	.596
Number of frames	124
watertight compartments	269

The keel of the *West Virginia* was laid on September 16, 1901, and that of the *Maryland* on October the 29th, of the same year, and, with the exception of the change of one stateroom on the *Maryland* to an office for the chief of staff, the only difference between the two ships is the name on the stern. The hull is built of basic, open-hearth steel, with the frames spaced 4 feet apart, except under the machinery and boilers. The inner bottom extends from frame 20 to frame 102, compartments B 78, 79, 82, 83, 86, 87, 90 and 91 and are arranged

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for reserve-feed water tanks, and are situated between frames 40 and 53, and are those on each side of the vertical keel.

A cofferdam extends the whole length of the ship on the protective deck; between frames 40 and 92 it is carried above the berth-deck level, and is filled with cornpith cellulose both forward and aft of these frames only.

Practically all of the wood work that is not fairly portable and can be dispensed with in time of war or thrown overboard before going into action is fire-proofed, except the teak armor backing, the wood work of the chart and pilot house, and the wood work below the protective deck.

The bridge deck and main decks are laid with wood, the others being covered with linoleum.

There are two military masts, each with three tops for machine guns on the lower two and searchlights on the upper.

THE BRIDGE DECK.

The bridge deck extends between the two turrets and forms the upper part of the superstructure. Over the forward end of the bridge deck is built the pilot house, chart room and emergency cabin, and above this is the flying bridge. Over the after end is the signal bridge, at the level of the top of the signal tower.

The following boats are carried on the bridge deck:

1 50-foot picket boat.
1 36-foot steam cutter.
1 33-foot steam cutter.
2 36-foot launches.
5 30-foot cutters.
1 30-foot barge.
2 30-foot whale boats.
1 30-foot whale-boat gig.
2 20-foot dinghys.
1 16-foot dinghy.
1 14-foot dinghy.
1 12-foot punt.
2 Carley life rafts.

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These boats are handled by one 15-ton, one 10-ton and two 3-ton electric cranes.

THE MAIN DECK.

This deck extends from stem to stern, and is partially covered by the bridge deck between the two turrets. The hawse pipes extend through the bow to this deck, and protecting plates are worked in to carry the chains to the windlass house, which is partly on this deck.

The two 8-inch gun turrets, the four 6-inch guns in separate casemates, and eight 3-inch R. F. guns are mounted on this deck. The 6-inch and 3-inch guns are covered by the bridge deck.

In the forward part of the superstructure is the bakery, the crew's galley, the Admiral's and Captain's galley, and the officers' galley. Abaft of the fireroom hatch are four rooms: the two forward are the printing office and wireless-telegraphy room; the two after ones are staterooms for two warrant officers. Between the two after 6-inch gun casemates are the warrant officers' mess room and pantry, and abaft the 'thwartship armor are situated the executive officer's stateroom, bath and office, the paymaster's office, the navigator's office and the ordnance officer's office.

THE GUN DECK.

From forward to aft on this deck comes the crew's shower bath, lavatory and head, windlass house; the casemate, containing the ten 6-inch guns, six warrant officers' staterooms, the staterooms of the senior engineer officer, navigator, paymaster, surgeon and flag lieutenant; the captain's cabin, stateroom, bath and office; the chief of staff's cabin and bath, and the quarters for the Admiral. Within the casemates are ten 6-inch B.L. rifles, separated by splinter bulkheads, the armory, the engineer's office, the mine room, and the general mess pantry.

On the *Maryland* the office of the chief of staff takes the place of the flag lieutenant's stateroom on the *West Virginia.*

THE BERTH DECK.

The after berth deck, from the 'thwartship armor, contains the junior officers and ward-room pantries, the mess room and

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twelve staterooms for junior officers; the ward-room mess room, extending the width of the vessel, with two circular and two large square air ports on each side; sixteen staterooms for ward-room officers, in two compartments, and the ward-room lavatory.

The forward berth deck contains the paints and oils, the brig, the contagious ward, the chief petty officer's quarters, staterooms for the master-at-arms and the sergeant of marines, refrigerating rooms, ice machine, laundry, drying room and sick bay, with operating room, dispensary and bath.

The sick bay is the most commodious and the best equipped of any heretofore installed in any vessel of the United States Navy, situated about one-fourth of the ship's length from the bow, having a clear space of 8 feet 9 inches under the beams, being well lighted and ventilated and having ample room in bunks for about five per cent. of the ship's crew. The contagious ward has four additional bunks. The operating room is adjacent to the sick bay and is well lighted and commodious, the deck is tiled and is easily cleaned.

THE ARMOR.

The armor consists of the side belt of Kruppized nickel-steel, 6 inches thick abreast of the machinery space and 3½ inches thick forward and aft of this. Above the belt armor is the lower casemate armor of 5 inches, the upper casemate of 5 inches and the superstructure armor of two inches, except around the 6-inch guns, where it is 5 inches.

The conning tower is 9 inches and the signal tower is 5 inches thick.

There are two elliptical balanced turrets, revolving over circular barbettes. The turret armor is 6½ inches on the inclined face, in which the gun ports are cut, and 6 inches on the vertical; the barbette armor is 6 inches. All 3-inch R.F. guns are protected by 2 inches of armor.

The casemate protection at the forward and after ends is 4 inches from the bridge deck to the protective deck.

The protective deck consists of two 30-pound plates, and this is further protected by one 100-pound plate on the slopes.

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ORDNANCE.

Main battery:

Four 8-inch 40-caliber breech-loading rifles.
Fourteen 6-inch 50-caliber breech-loading rifles.

Secondary battery:

Eighteen 3-inch 50-caliber rapid-fire rifles.
Twelve 3-pounder rapid-fire rifles.
Four 1-pounder automatic.
Four 1-pounder Hotchkiss rapid-fire rifles.
Six 30-caliber Colt's automatic.
Two Gatling guns.

THE MAIN ENGINES.

There are two four-cylinder triple-expansion engines of the vertical, inverted, direct-acting type, in separate watertight compartments. The order of the cylinders is: the forward L.P., the H.P., I.P and after L.P. The H.P. and I.P. cranks are at ninety degrees, and the two L.P. cranks at the same angles. The H.P. and F.L.P. cranks are opposite.

The cylinders are supported by twelve forged, nickel-steel, cylindrical columns which are tied fore and aft by heavy cast-steel I braces and diagonal rods and strengthed 'thwartships by X frames bolted to the columns at top and bottom and stayed by through tie bolts and distance pieces at center of X.

All valves of the single-ported piston type, of cast-iron heads, with cast-iron packing ring and steel follower. The I.P. and L.P. valves take steam from the outside and the H.P. valves from the inside. The valves are arranged to lift through the upper head. The H.P. cylinder has one valve, and there are two for each I.P. and L.P. cylinder. Each valve is provided with a balance piston working in cylinder cast with the upper valve chest covers.

The valve gear is of the Stephenson type with double-bar links. The H.P. valve stem connects with its link block direct, while the I.P. and L.P. stems are connected by means of a crosshead.

The engines are operated by a floating-lever, steam reversing

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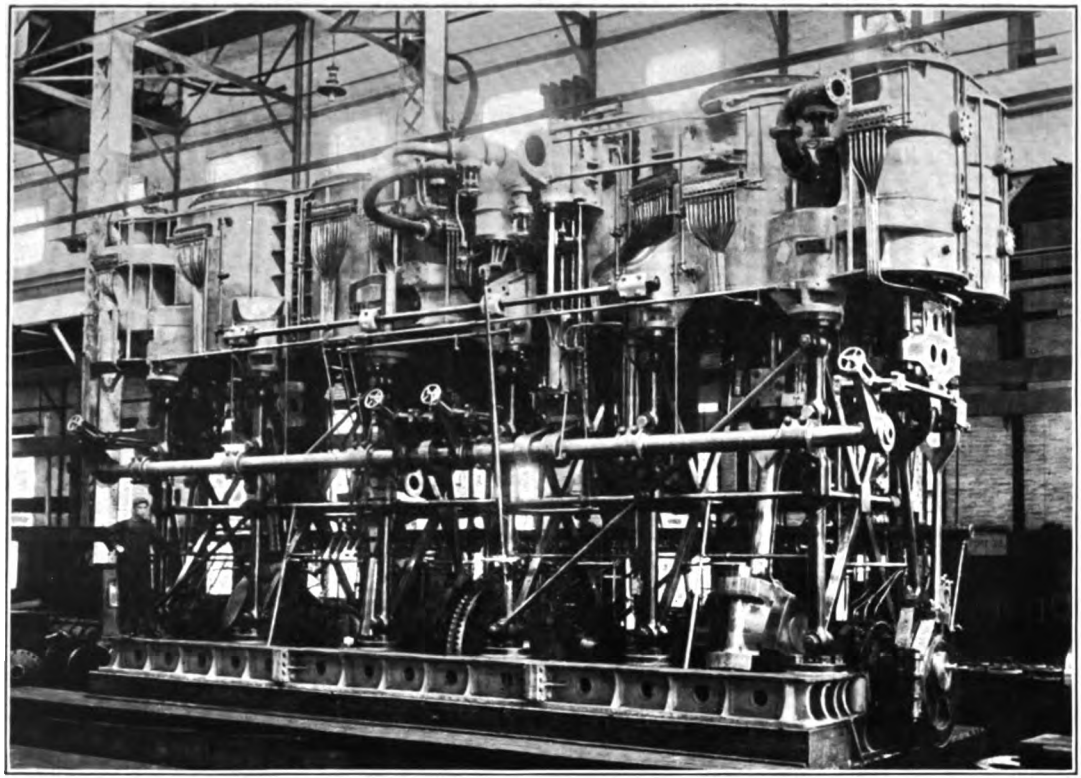

One Main Engine, U. S. S. "West Virginia."

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cylinder whose piston rod carries a piston through an oil-control cylinder and is connected to the reverse shaft supported on the engine columns. This reverse shaft carries the reverse arms, in which independent adjustable cut-off blocks are fitted to connect to each link, giving a range of cut-off from .5 to about .73 of the stroke. When the H.P. crank is on the center live steam may be admitted into the I.P. and L.P. receivers.

All the cylinders are jacketed around the working liners and at both ends. The live steam at working pressure enters the high-pressure jackets, and from thence by reducing valves goes to the I.P. and L.P. jackets. Each engine has a 13-inch throttle valve, balanced by a piston working in the cylinder cast with the valve casing.

The H.P. pistons are of cast iron and the others are of cast steel, conical in shape; the followers are of cast-steel, the H.P. and I.P. packing rings are solid cast-iron rings kept central by springs, and the L.P. piston packing is two rings of segments set out by steel springs. The piston rods are hollow, of high-grade nickel-steel, oil tempered and annealed. They are accurately ground, cylindrical and polished, tapered at each end to fit the piston and crosshead. The crosshead is of nickel-steel, forged with the crosshead pins, and is bolted to the cast-steel slipper. The go-ahead guides are of cast-iron, hollow for water circulation, and are supported by being bolted to lugs on the cylinders at the top and to the cast-steel brace beam at the bottom. The backing guides are of cast steel and are bolted to flanges on the ahead guides.

The connecting rods are of the same material as the piston rods, are hollow, are forked at the upper end to carry the crosshead brasses. The crank-pin brasses are of composition and are held to the stub end of the connecting rod by a steel cap and through bolted.

The eccentrics are in two parts, the larger of cast iron and the smaller of wrought steel. The eccentric straps are of composition, the rods of high-grade nickel-steel.

The turning engine is a double, inverted engine, secured to

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the engine bed plate and connected by worm gearing to the crank shaft.

The working platforms are inboard; the reversing lever, the throttle wheel, and engine telegraph are placed within easy reach of each other; the H.P. valve-chest steam gauge is at the throttle. The clock is on the center-line bulkhead over the engine-room desk, and directly beside it are the counters, worked by gears and shafting from the thrust shaft.

ENGINE DATA.

Cylinders, number for each engine	4
H.P., diameter, inches	38.5
I.P., diameter, inches	63.5
L.P., diameter, inches	74
Stroke of all pistons, inches	48
Valves, H.P. (one for each cylinder), diameter, inches	23
I.P. (two for each cylinder), diameter, inches	25.5
L.P. (two for each cylinder), diameter, inches	28.5
Valve stems, forged nickel-steel, diameter, inches, H.P. $3\frac{3}{8}$, I.P. $3\frac{1}{4}$, L.P.,	$3\frac{1}{4}$
diameter through valve, inches	$2\frac{1}{4}$
at balance pistons, inches	$1\frac{3}{4}$
Main steam pipe, diameter at throttle, inches	13
area of cross section, square inches	132.73
Volume swept by H.P. piston, mean, per stroke, cubic feet	31.55
I.P. piston, mean, per stroke, cubic feet	87.18
both L.P. pistons, mean, per stroke, cubic feet	237.36
Net area of I.P. to H.P. piston	2.76
both L.P. to H.P. pistons	2.72
Piston rods, diameter, inches	$8\frac{1}{2}$
of axial holes, inches, H.P. and I.P., $4\frac{1}{2}$; L.P.,	5
length, feet and inches	$9-9\frac{1}{8}$
Cylinder walls of H.P., I.P. and L.P. cylinders, thickness, inches	$2\frac{1}{4}$, $1\frac{7}{8}$, $1\frac{3}{4}$
liners of cylinders, thickness, inches, H.P. and I.P., $1\frac{3}{4}$; L.P.,	$1\frac{1}{2}$
Valve-chest liners, thickness, inches	$1\frac{3}{8}$
Jacket space, inch	$\frac{3}{4}$
Cylinder relief valves (two on each cylinder), diameter, inches	$4\frac{1}{2}$
Connecting rod, length, center to center, inches	96
diameter, upper end, inches	$8\frac{1}{4}$
lower end, inches	$9\frac{1}{4}$
axial hole, inches, H.P. and I.P., 5; L.P.,	$5\frac{1}{2}$
crosshead bolts (four), diameter, inches	$3\frac{1}{2}$
crank pin bolts (two), diameter, inches	$4\frac{3}{4}$
Crosshead, surface, ahead, square inches, H.P. and I.P., 717.1; L.P.,	525.3
backing, square inches, H.P. and I.P., 230.8; L.P.,	294.7
pin, diameter, inches	$10\frac{3}{4}$

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Crosshead, diameter of axial hole, double tapered, inches	7 to 2
length of pin, inches	11
Crank shaft, number of sections	2
diameter, inches	18½
of axial holes, inches	10
coupling discs, inches	32
thickness coupling discs, inches	4½
coupling bolts in one flange (twelve), diameter, inches	3¼
journals, diameter, inches	18½
forward to aft length, inches, 4 of 22½, 1 of 35 and 1 of	29½
length of forward sections, feet and inches	16–8½
length of after sections, feet and inches	18–7½
maximum distance between journals, feet and inches	6–5 9/16
pin, diameter, inches	20
length, inches	23
projected area, square inches	460
diameter, axial holes, inches	12
webs, width, inches	22¼
thickness, inches	11
Thrust shaft, diameter, inches	17¾
axial hole, inches	10½
collars, number on each shaft	12
diameter, inches	27
thickness, inches	2
space between, inches	4
total thrust surface both engines, sq. inches	7,802.6
length, feet	13
steady bearings, inches	26
Line shaft, diameter, inches	17½
axial hole, inches	10½
coupling discs, inches	32
coupling bolts in one flange (twelve), diameter, inches	3¼
length, feet and inches	23–3
Propeller shaft, without casing, diameter in strut bearing, inches	forward 18 17/32; aft, 18½
length, feet and inches	47–11¾
Stern-tube shaft, after stern-tube bearing, inches	18½
forward stern tube bearing, inches	18 9/16
length, feet and inches	39–8
Inboard collar coupling, diameter, inches	33½
length, inches	22
of collar, inches	14
Stern-bracket bearing, diameter, inches	forward, 20 11/16; aft, 20¾
length, inches	forward, 35¾; aft, 59¼
Reversing engine, diameter of steam cylinder, inches	16
controlling cylinder, inches	8
stroke, inches	22
Turning engine, diameter of steam cylinders (two), inches	7
stroke of pistons, inches	5

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West Virginia:	*Star.*	*Port.*	*Star.*	*Port.*
Clearance of H.P. cylinder, per ct., top,	25.22,	21.91;	bottom, 27.81,	29.01
I.P. cylinder, per ct., top,	20.06,	20.51;	bottom, 22.90,	22.52
F.L.P. cylinder, per ct., top,	16.62,	16.73;	bottom, 19.62,	19.19
A.L.P. cylinder, per ct., top,	16.17,	16.34;	bottom, 19.84,	19.05
Maryland:				
Clearance, H.P. cylinder, per cent., top,	24.18,	26.77;	bottom, 29.55,	27.38
I.P. cylinder, per cent., top,	20.82,	20.13;	bottom, 24.14,	23.37
F.L.P. cylinder, per ct., top,	15.72,	15.95;	bottom, 19.65,	19.28
A.L.P. cylinder, per ct., top,	16.39,	16.17;	bottom, 19.79,	19.62

MAIN CONDENSERS.

There is one main condenser in each engine room. The shell is of steel, the water heads of composition, the tubes are of Admiralty metal, not tinned. There is a by-pass in the division plate of the water chest at the injection and delivery end, to allow the circulating pump to deliver directly overboard when working on the bilge, injection or the main drain.

Diameter of shell (inside), feet and inches	7- 5
Thickness of shell, inch	$\frac{7}{16}$
Length over heads, feet and inches	16-10
Tubes, diameter (outside), inch	$\frac{5}{8}$
length between tube sheets, feet and inches	14- 0
thickness, No. 18, B.W.G., inch	.049
spacing between centers, inch	$\frac{15}{16}$
number in each condenser	6,292
Condensers (two), cooling surface, square feet, each	14,411

MAIN AIR PUMPS.

There is located abaft the condenser in each engine room one Blake, two-cylinder, vertical, double-acting air pump. There are suction valves below the pump piston, and these are reached by removing large man-hole covers that form a part of the pump barrel. The valves in the pump piston may be reached in the same manner when the piston is at the bottom end of its stroke.

Diameter of cylinders, inches, each	14
pump cylinders (two), inches	35
Stroke, inches	18
Diameter of piston rods (nickel steel), inches	$2\frac{3}{4}$
pump rods (Tobin bronze), inches	$3\frac{1}{2}$
pump valves in bucket, inches	$5\frac{1}{2}$
Number of delivery valves in each end	18
Area through 18 valve openings, square inches	234
by lift, square inches	185.55

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MAIN CIRCULATING PUMPS.

There is one centrifugal, double-inlet, circulating pump driven by a two-cylinder, compound, vertical, inverted engine located forward of the main condenser. Each pump has a capacity of 15,000 gallons of water when its engines are run at 266 revolutions per minute. Each pump draws from the sea, the main drain and from the engine-room bilge, and discharges through the condenser or directly overboard, through the condenser by-pass, to the overboard delivery. The suction valves to these pumps are so locked that the sea valve and the bilge or drain valve cannot be open at the same moment.

Diameter of steam cylinders (two), inches	H.P. 11, L.P. 22
Stroke, inches	10
Diameter of pump runner, inches	45
Width of pump runner at periphery, inches	4
hub, inches	$10\frac{9}{16}$
Diameter of inlet and outlet nozzles, inches	two $15\frac{1}{2}$ and one 21

FEED AND FILTER TANKS.

There is in each engine room, under and extending to the air pump, a feed tank. Over the after end of the feed tank, and combined with it, is a filter tank of 6,500 gallons capacity for both tanks. The discharge from the main air pump, from the auxiliary condensers and from the dynamo condenser, on the port side, enters the first compartment of the filter tank, passing through loose bagged coke into the second compartment, where the water is again filtered through the same material before reaching the third compartment, where it is finally filtered before entering the feed tank.

A valve is fitted in the feed tank which is operated by a copper float to prevent air from entering the feed pipes. This valve may be operated by a lever on the outside of the tank. Each compartment in the filter tank is provided with a vapor pipe.

FEED PUMPS.

There are two main feed pumps in each engine room which draw water direct from the feed tanks or from the discharge of the hot-well pumps through the feed heaters. These pumps

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have no other suction, and discharge into the main feed line only.

The auxiliary feed pumps are six in number and are located on the center-line bulkhead, one in each double fireroom. They draw from the feed tank, the sea, the boiler blow pipes, the main and auxiliary drains and the bilges in their respective compartments, and discharge into the auxiliary feed line, overboard or to the fire main.

Main feed pumps (four): Blake, simplex, vertical, center-packed plunger.	
Diameter of steam cylinders, inches	14
water cylinders, inches	10
piston rods, inches	2¾
Stroke, inches	18
Auxiliary feed pumps (six):	
Diameter of steam cylinders, inches	13
water cylinders, inches	9½
piston rods, inches	2½
Stroke, inches	12

FEED-WATER HEATERS.

There is one feed-water heater in each engine room containing 950 square feet of heating surface. There are 852 tubes in each heater. Steam from the auxiliary exhaust line is bypassed into the heater and the back pressure is regulated by a spring relief valve. The hot-well pump takes water from the feed tank and delivers it through the tubes of the feed heater to the main feed-pump suction.

HOT-WELL PUMPS.

There is one hot-well pump in each engine room. This pump can drain water from the feed tank, the reserve double-bottom compartments and from the connections in the ship's side, where a hose may be connected for pumping out a water boat. The discharge is through the feed heater to the feed-pump suction, to the feed tank and also to the reserve tanks.

FIRE AND BILGE PUMP.

There is one fire and bilge pump in each engine room. These pumps draw from the sea, the drainage system and

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from their respective bilges. They discharge into the fireroom or overboard.

Number and type of pumps: two F. and B. Blake, simplex, vertical, piston.

Diameter of steam cylinders, inches	12
water cylinders, inches	10
piston rods, inches	$2\frac{1}{4}$
Stroke, inches	12

DISTILLER CIRCULATING PUMP.

There is one distiller circulating pump located in the evaporator room. This pump draws from the sea and delivers overboard or to the flushing main through the distillers or directly. By means of connecting valves the flushing main may be served by any pump discharging to the fire main.

AUXILIARY CONDENSERS.

There is one auxiliary condenser in each engine room and one in the lower dynamo room on the port side. These are of the well known Wheeler type, with air and circulating pump attached to and under the condenser.

FORCED-DRAFT BLOWERS.

There are sixteen Sturtevant blowers situated on the protective deck over the fireroom hatches and deliver directly into the boiler compartments. These blowers are encased, and from their excellent location give a most efficient service. They may be easily cared for and overhauled with ease, thus ensuring a long life and immediate readiness for service at all times.

FORCED DRAFT.

Number of blowers	16
Type of blower engines	Two-cylinder, vertical, double enclosed.
Seam cylinders (2) diameter, inches	6
Stroke, inches	5
Diameter of piston rods, inches	$1\frac{13}{16}$
fan, inches	66
Area of induction nozzle, square inches	1,385.45
eduction nozzle, square inches	1,728

THE GENERAL WORKSHOP.

Located on the berth deck over the evaporator room and extending athwartships to the width of the engine-room

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hatches is the general worshop. The tools are driven by one ten-horsepower electric motor, and the usual counter shafting, belting and pulleys restrict the already limited head room.

The following tools are installed:

One 30-inch screw-cutting back-geared gap lathe, 10 feet between centers.

One 14-inch screw-cutting back-geared lathe, 4 feet between centers.

One column-shaping machine of 16-inch stroke and 26-inch traverse.

One back-geared drilling machine to drill up to 1½ inches and 14 inches from edge of work.

One 16-inch sensitive drill.

One No. 1 Universal milling machine.

One combined hand punch and shears,

One double-wheel emery grinder.

One 30-inch grindstone.

Six bench vices.

ASH HOISTS.

There are five ash-hoist engines, made by the Williamson Bros., of the usual servo-motor type, and are located in the fireroom hatches on the level of the gun deck. The buckets are hoisted through the ventilators to the main deck and trolleyed to the ash chutes at the sides.

DISTILLING APPARATUS.

There are four evaporators, arranged to be run in two pairs for double effect or each in single effect. There are two horizontal distillers in the evaporator room so arranged that the steam goes through the tubes. The rated capacity of the evaporators is 23,000 gallons of water per diem and that of the distillers is 10,000 gallons. This was met on the test by running all four evaporators in two double-effect pairs and distilling the water from the second-effect evaporator. This distilled water was measured, and exceeded the rated capacity of the distillers on a six-hour test.

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ICE MACHINE.

There is supplied one 3-ton Allen dense-air machine. This is of the latest improved design, and showed excellent results on the tests. From the time of starting the machine, everything being at atmospheric temperature, the pressures were established in 45 minutes, one full batch of ice (135 pounds) was made in 3½ hours and the temperatures of the cold-storage rooms were reduced from 70 to 8, 14, 18 and 24 degrees, respectively, in six hours. In addition to this two scuttle butts, one on the deck above and one on two decks above, were cooled to 40 degrees.

ANCHOR WINDLASS.

There is a vertical Hyde steam windlass on the gun deck forward, the engine being located on the deck outside of the windlass house.

STEERING ENGINE

There is a Williamson Bros. combined hand-and-steam steering engine. The engine is controlled from the bridge, pilot-house and conning tower by a wire rope actuating the engine-valve and by a hydraulic telemotor to accomplish the same purpose.

TORPEDO OUTFIT AND AIR COMPRESSORS.

At the same time of writing this article the torpedo plant has not been fully settled nor installed. The vessels are to be provided with two submerged torpedo tubes, with the necessary compressors, and in connection with the compressors an additional air compressor will be installed to supply the smoke-blowing outfit.

SCREW PROPELLERS.

The propellers are of manganese-bronze and turn inboard when going ahead, the starboard propeller being left-handed. They are designed as true screws, and the surfaces of the blades are ground smooth and tinned for the *West Virginia* and not tinned for the *Maryland*. In smoothing the blades, no attempt was made to grind off the inequalities of the casting nor to fair the blade to a true helicoidal surface. On the run of the *West Virginia* to Boston it was found that the port

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engine averaged about 3 to 4 revolutions less than the starboard, and the vessel carried a port helm of about one degree. On the speed trial the average difference was 4.45 revolutions per minute. There was also a difference in the horsepower, the port engine developing 1,502.7 I.H.P. more than the starboard. This was due to the use of steam in the P.L.P. receivers in order to try and bring up the revolutions of the port engine to those of the starboard.

On the return from the trial to the shipyard the vessel was docked and the pitch of the screws remeasured. It was found that the port screw showed about six inches more pitch than the starboard, and this may have been due to faulty measuring, or the blades may have been sprung when first backing away from the dock by catching a heavy rope cable around the port screw.

Taking the average revolutions on the trial and the pitch as remeasured, the starboard engine drove the vessel 2,118 feet more in an hour than the port (this is neglecting the slip of both wheels), and this accounts for the port helm. The port wheel was reset to correspond with the starboard.

In order to avoid any difference in the two wheels of the *Maryland* and to set each blade to correspond with all others, the question of measuring the pitch and setting the blades was taken up personally by the author, and he, in company with Mr. A. A. Howitz, one of the leading engine draftsmen of the company, very carefully set each blade at a pitch of 22 feet and 6 inches at a radius of 6 feet 6 inches. The pitchometer used was somewhat crude, but of sufficient accuracy to make all errors constant, as was tested by remeasuring a blade that had been removed from the hub and replaced in the marks, when it was found that the pitch was the same as before. The contractors remeasured the blades by laying the propellers on a planer bed and measuring from the horizontal surface. These measurements showed the port propeller to be as before set, but that the starboard propeller was not as high. The pitch of this wheel was increased to bring it up to the port wheel as measured when on the bed plate, and remeasured by

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means of a pitchometer for use when the plane of rotation is horizontal. After a dock trial, it was claimed that the starboard engine ran faster, with less power, than the port, and the contractors reset the starboard propeller to agree with certain shop marks furnished from the pattern, and the pitch of both wheels was reported to the Board of Inspection to be 22 feet 6 inches.

Number of blades	3
Diameter, feet and inches	17- 3
of hub, inches	54
Length of hub, inches	42
on official trial, *West Virginia*, mean, ft. and ins.	22-044
Maryland, mean, ft. and ins.	22- 6
Helicoidal area, square feet	83.7
Disc area, square feet	240.53
Pitch ÷ diameter, *West Virginia*	1.275
Maryland	1.304

ELECTRIC PLANT.

The electric plant of these vessels (and their class) is the largest of any heretofore installed in any vessel of the United States Navy.

There are three 100-kilowatt and four 50-kilowatt generating sets, all at a pressure of 125 volts.

All of the auxiliary machinery in the different compartments outside of the engine and fire rooms, except the anchor windlass, the forced-draft blowers, and the steering engine, is actuated by electric motors.

The dynamo rooms are immediately forward of the boiler compartments, and the generators take steam directly from the auxiliary steam pipes.

The lighting output supplies 1,200 incandescent 16-candlepower lamps, 1,329 incandescent 5-candlepower lamps for use in the battery-control instruments, 14 5-ampère arc light (about 500-candle power each), one in each fireroom and two in each engine room. There are 6 30-inch hand-controlled searchlights of 21,000 candlepower each.

There are in all 103 separate electric motors, the principal ones being—

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		Horsepower.
2	Torpedo air-compressor motors (40 each),	80
2	Boat-crane motors (35 each),	70
7	Deck-winch motors (25 each),	175
2	Boat-crane motors (20 each),	40
2	Turret-turning motors (15 each),	30
1	Workshop motor (10),	10

In addition to the above there are 46 small fan motors, two night-signalling sets, and 6 portable ventilating sets.

The total horsepower of the electric plant is about 670 I.H.P., or 2⅛ per cent. of the total horsepower at full speed.

DYNAMO ENGINES.

Type	Cross compound
50-kw. dynamo engines, number	4
Diameter of cylinder, H.P., inches	7½
L.P., inches	14
Stroke, inches	8
Diameter of piston rods, H.P., inches	1¾
L.P., inches	1¾
100-kw. dynamo engines, number	3
Diameter of cylinders, H.P., inches	10
L.P., inches	18
Stroke, inches	10
Diameter of piston rods, H.P., inches	2¼
L.P., inches	2¼

BOILERS.

There are sixteen Babcock and Wilcox boilers of the latest improved type. The arrangement of drums, baffling and circulation, is shown in the cut, reproduced from an article by Lieutenant R. K. Crank in the previous issue of the JOURNAL, and to which article the reader is referred for a full description of the boilers of this vessel:

BOILER DATA.

Floor space, length, feet and inches	9– 1½
width, feet and inches	16– ½
Length of grate, feet and inches	7– 0
Width of grate, feet and inches	7– 1½
Grate surface, one boiler, square feet	100
all boilers, square feet	1,600
Heating surface, one boiler, square feet	4,434

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Heating surface, all boilers, square feet	70,944
to grate surface	44.34
Number of tubes in one boiler (2-inch, 901; 4-inch, 57)	923
elements, one boiler	27
Weight of water in one boiler at steaming level, pounds	15,750
one boiler complete, pounds	89,068

VENTILATION.

On these ships all the spaces below the gun deck, and the officers' and crew's closets on the gun deck, are provided with artificial ventilation. There are sixteen systems (or units) of ventilation, and these are grouped into four methods, viz:

The supply and exhaust method.—This method is applied to all compartments where it is difficult both to supply the fresh air and to exhaust the foul by natural convection. Under this head come the dynamo rooms and steering-engine room.

The supply method.—Air is supplied by forced ventilation to all compartments below the gun deck and between the 'thwartship armor hulkheads, and also, where necessary, as in the magazines, exhaust ducts are provided to allow the escape of the foul air.

The exhaust method.—Where there is a natural inlet of fresh air through hatches and air ports, as in the case of living quarters forward and aft on the berth deck, the foul air is exhausted by fans.

The convection method.—This is applied to the bunkers and firerooms where there is a natural flow of heated air upward through hatches and pipes provided for that purpose and ventilators and supply pipes are arranged to carry the fresh air to supply the need.

The engine-room ventilation is very successfully planned. There are four electric blowers located on the gun deck and supply fresh air from ventilators to the four large ducts leading inboard and outboard in each engine room. From these ducts branches are led to the various points in the engine rooms where cool and fresh air is most needed, and from the various tests it is believed that a fairly cool and even temperature will be kept even in the hottest climate.

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THE SPEED TRIAL OF THE *WEST VIRGINIA*.

The *West Virginia* left her anchorage off the shipyard at Newport News at 12·05 P. M. of October 29, 1904, having on board 517 officials, observers and crew. Of this number 315 were in the Engineer's department, 109 in the chief steward's department, 10 representatives of sub-contractors and 25 representatives of the Government, including enlisted men who were to serve on this vessel in commission.

There was on board 1,960 tons of coal (402 in bags), 5,495 gallons of oil and 958.35 tons of fresh water. The run to Boston was made without mishap and the vessel was anchored in President's Roads early on the morning of the 31st.

On the morning of November 2 the start was made for the Cape Ann trial course, and the vessel was run over the course and return with the following performance:

SYNOPSIS OF STEAM LOG.

West Virginia.

Steam pressure at boilers (gauge), pounds	274.1
H.P. steam chest, S. engine (gauge), pounds	236
P. engine (gauge), pounds	236
first receiver, S. engine, absolute, pounds	99
P. engine, absolute, pounds	104.2
second receiver, S. engine, absolute, pounds	47
P. engine, absolute, pounds	50.9
Vacuum in condensers, in inches of mercury, port, 26.5; starboard,	26
Mean effective pressures in cylinders, in pounds, per square inch:	
Main engines, starboard, H.P	107.8
I.P	32.56
F.L.P	18.04
A.L.P	19.56
port, H.P	107.3
I.P	32.53
F.L.P	26.3
A.L.P	26.27
Mean pressure in pounds per square inch on L.P. piston, equivalent to aggregate M.E.P. on all pistons, starboard	44.96
Mean pressure, in pounds per square inch, on L.P. piston, equivalent to aggregate M.E.P. on all pistons, port	54.2
Revolutions per minute:	
Main engines starboard, 129.7; port,	125.25
Pumps, air starboard, 35.6; port,	27.3
circulating starboard, 164.6; port,	177.35
Main feed, double strokes per minute starboard, 20.8; port,	27.3
Pumps, auxiliary, fire and bilge starboard, 46.1; port,	46.5

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Speed of ship in knots per hour	22.146
Slip of propeller in per cent. of its own speed, based on mean pitch.....starboard, 21.35; port,	18.56
Air pressure in firerooms, in inches of water	1.87
Indicated horsepower:	
Main engines, starboard H.P.	3,847.8
I.P.	3,213.5
F.L.P.	2,423.1
A.L.P.	2,627.3
total	12,111.7
port, H.P.	3,694.1
I.P.	3,101.8
F.L.P.	3,412.7
A.L.P.	3,408.8
total	13,614.4
Air pump.....starboard engine, 23.46; port engine,	15.06
Circulating pump.....starboard engine, 52.2; port engine,	68.6
Feed pumps, main (four)	137.52
auxiliary (six)	83.65
Fire and bilge pumps (two)	10.77
Hotwell pumps (two)	28.24
Distiller circulating pump (estimated)	14
Dynamo auxiliary-condenser circulating pumps	2.034
Blower engines for forced draft in firerooms (sixteen)	219.84
Dynamo engines	86.13
Collective of both main engines	25,726.1
main engines, air, circulating, hot-well and feed pumps	26,134.83
main and auxiliary engines in operation during trial	26,465.57
all machinery during trial, per square foot of G. S.	16.54
H. S.	.373
Main engines, air, circulating and feed pumps, per square foot of G. S.	16.33
Main engines, air, circulating and feed pumps, per square foot of H. S.	.368
Coal: kind and quality.....Bagged hand-picked New River.	
pounds per hour	80,389
pounds per hour per I.H.P. collective, all machinery in operation	3.037
pounds per hour per square foot of G. S.	50.24
H. S.	1.12
Cooling surface, square feet per I.H.P.	1 08
Heating surface, square feet per I.H.P.	2.68

PUMPS IN THE ENGINE ROOMS.

Number and type of pumps.....Two hot-well, Blake simplex, vertical piston.	
Diameter of steam cylinders, inches	12
water cylinders, inches	16

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Diameter of piston rods, inches	2¼
Stroke, inches	16

AUXILIARY CONDENSERS.

	(2) *Engine Room.*	*Dynamo.*
Cooling surface, each, square feet	601.4	1,200.9
Diameter of steam cylinders of pump, inches	8	12
circulating pump cylinder, inches	10	14
air-pump cylinder, inches	10	14
Stroke, common, inches	12	12
Air pump, kind of valves	Hard rubber.	

After the test of the helm the speed was slacked until the steam pressure was under control, and the following practical trials were made: (*a*) With the engine stopped, the vessel having been brought to a dead still position, head to wind and helm amidship, the starboard engine was started at full speed astern and the port engine at full speed ahead. The time of turning through 360 degrees was 14 minutes and 23 seconds. (*b*) The same trial was made with the engines reversed: time, 13 minutes and 37 seconds.

This compares most favorably with the *Colorado*, which is fitted with out-turning screws, and occupied 15 minutes 51⅖ seconds on trial (*a*) and 15 minutes on trial (*b*).

The coal used on the run to and from Boston was estimated at 520 tons each way. There were 4,335 gallons of oil used on the whole round trip, and 1,300 pounds of tallow. Of the fresh water carried, 211 tons were used on trip to Boston—65.7 tons on the trial run and 226.3 tons on the return trip. This does not include the water used in Boston, nor does it include any water used for other than engine and boilers. An increase of reserve-feed supply from 150 tons to 275 tons was recommended by the Trial Board.

TRIAL OF U. S. S. *MARYLAND*.

The *Maryland* left Newport News at 11·25 A. M., on January 21st, 1905, and arrived at Boston, Mass., at 12·43 A. M., on January 23d.

The draught on leaving was taken before heaving in, and was 24 feet 4½ inches forward, and was 26 feet 4 inches aft.

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In order to obtain better results on this vessel than her sister, the contractors installed an electric device for signalling the periods of firing the furnaces. This device consisted of a motor and gearing arranged to make a contact on a lighting circuit at regular intervals. The gears gave six different changes, and by regulating the speed of the motor any desired interval could be obtained. The firemen were, with few exceptions, brought from New York just before the ship left Newport News, and showed little or no training in handling fires to produce economy in consumption of coal. The regularity of firing, however, showed its results in the ability to carry a higher pressure of steam, in a greater horsepower and increased speed.

In order to obtain data for working the machinery on the official trial the following tests were made on the run north:

TEST A.—NO STEAM IN RECEIVERS.

	Starboard.		*Port.*	
Revolutions per minute	119		118	
Main steam	270		273	
Vacuum	28.2		26.1	
H.P. valve chest	245		233	
First receiver	80		81	
Second receiver	18		15.5	
I.H.P., H.P. cylinders	3,810		3,480	
L.P. cylinders	4,075		3,980	
F.L.P. cylinders	1,817	3,877	1,747	3,522
A.L.P. cylinders	2,060		1,875	
	11,762		11,082	
Total I.H.P. (engines only)				22,844

TEST B.—STEAM FIRST RECEIVER ONLY.

	Starboard.		*Port.*	
Revolutions per minute	126		126	
Main steam (gauge)	268		270	
Vacuum	26		26	
H.P. valve chest	257		257	
First receiver	105		108	
Second receiver	25		22	
I.H.P., H.P. cylinders	3,605		3,655	
I.P. cylinders	5,300		5,275	
F.L.P. cylinders	2,475	4,963	2,180	4,690
A.L.P. cylinders	2,488		2,510	
Total I.H.P.				27,488

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TEST C.—STEAM IN FIRST AND SECOND RECEIVERS.

	Starboard.		Port.	
Revolutions per miuute	126		126	
Main steam	249		250	
Vacuum	26.5		26.6	
H.P. valve chest	242		240	
First receiver	102		103	
Second receiver	28		29	
I.H.P., H.P. cylinders	3,440		3,272	
I.P. cylinders	4,810		4,730	
F.L.P. cylinders	2,576	5,364	2,892	5,982
A.L.P. cylinders	2,788		3,090	
Total				27,598

TEST D.—STEAM IN SECOND RECEIVER.

	Starboard.		Port.	
Revolutions per minute	123		122.5	
Main steam	257		265	
H.P. valve chest	248		245	
First receiver	85		85	
Second receiver	27		26	
I.H.P., H.P. cylinders	4,278		4,110	
I.P. cylinders	3,711		3,380	
F.L.P. cylinders	2,411	5,086	2,810	5,562
A.L.P. cylinders	2,675		2,752	
Total				26,117

Before the vessel left Newport News additional vacuum gauges were fitted by direction of the Bureau of Steam Engineering at various points in the exhaust line, condenser and air-pump suction. The following table gives the reading of these gauges during the different tests shown above:

READING OF VACUUM GAUGES.

	Test A.		Test B.	
	Star.	Port.	Star.	Port.
Air-pump suction	27.2	27.2	25.4	26.8
Bottom of condenser	25.6	27.2	25.8	25.8
Top of condenser	26.5	26.7	24.7	23.7
F.L.P. exhaust, forward end	28.1	25	23.8	21
after end	24.2	23.7	21.3	19.9
A.L.P. exhaust, after end	28	28	21.9	22
forward	24.8	24.5	21.9	20.6

SYNOPSIS OF STEAM LOG.

Steam pressure at boilers (gauge), pounds	278.2
H.P. steam chest, S. engine (gauge), pounds	250.5
P. engine (gauge), pounds	246.5

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Steam pressure at first receiver, S. engine (absolute), pounds.....	126.75
P. engine (absolute), pounds.....	122.
second receiver, S. engine (absolute), pounds...	48.5
P. engine (absolute), pounds..	47.5
Vacuum in condensers, in inches of mercury, port, 25.3; starboard,	25.8
Mean effective pressures in cylinders, in pounds, per square inch:	
Main engines, starboard H.P.....	89.06
I.P.....	42.
F.L.P.....	23.48
A.L.P.....	25.5
port H.P.....	81.19
I.P.....	45.19
F.L.P.....	24.72
A.L.P.....	25.23
Mean pressure in pounds per square inch on L.P. piston, equivalent to aggregate M.E.P. on all pistons, starboard.....	51.76
Mean pressure in pounds per square inch on L.P. piston, equivalent to aggregate M.E.P. on all pistons, port.....	52.361
Revolutions per minute:	
Main engines..... .....starboard, 127.72; port,	127.96
Pumps, air.....starboard, 27.36; port,	37.6
Circulating.....starboard, 174.5; port,	182.9
Pumps, main feed, double strokes per minute, starboard, inboard....	21.9
outboard..	23.9
port, inboard.....	23.
outboard.....	21.
auxiliary, fire and bilge.....starboard, 48.4; port,	47.5
Speed of ship in knots per hour.....	22.406
Slip of propeller in per cent. of its own speed, based on mean pitch..... .....starboard, 20.99; port,	21.15
Air pressure in firerooms, in inches of water.....	1.93
Indicated horsepower:	
Main engines, starboard H.P.....	3,131.6
I.P.....	4,081.7
F.L.P.....	3,106.
A.L.P.....	3,373.2
total.....	13,692.5
port H.P.....	2,860.
I.P.....	4,399.6
F.L.P.....	3,276.1
A.L.P.....	3,343.1
total.....	13,878.8
Air pump.....starboord engine, 16.12; port engine,	18.008
Circulating pump.....starboard, 66.09; port,	77.14
Feed pumps, main (four).....	174.
auxiliary (four).....	133.6
Fire and bilge pumps.....starboard, 2.2; port,	0.2
Hot-well pumps.....starboard, 9.95; port,	11.92

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Indicated horsepower:	
Circulating pump (estimated), flushing	4.5
Blower engines for forced draft in firerooms (sixteen)	352.
Dynamo engines (one)	56.57
Collective, both engines, main	27,571.3
main engines, air, circulating, hot-well and feed pumps	28,058.95
main engines and all auxiliaries in operation during trial	28,474.42
all machinery during trial, per square foot of G.S	17.79
main engines, air, circulating, hot-well and feed pumps, per square foot of G.S	17.536
main engines, air, circulating, hot-well and feed pumps per square foot of H.S	0.3955
Coal, kind and quality, bagged, hand-picked New River, pounds per hour	81,148.2
Pounds per hour per I.H.P. collective, all machinery in operation	2.849
Pounds per hour per square foot of G.S	50.74
H.S	1.144
Cooling surface, square foot per I.H.P	1.012
Heating surface, square foot per I.H.P	2.492

The *Maryland* carried 2,010 tons of coal, 687.38 tons of fresh water for steaming purposes, 5,298 gallons of oil and 287.4 tons of fresh water for ship's use.

On the run from Cape Charles lightship to Winter Quarter Shoal lightship, a distance of 57 knots, the ship showed:

Revolutions per minute	99.5
Steam (main gauge)	262.6
Vacuum	28.2
Speed (average)	17.58
Slip, per cent	20

Just after starting on the trial course the discharge air chamber of the port hot-well pump broke, and this pump had to be temporarily stopped. This necessitated the use of feed direct from the feed tank without passing through the feed heater. The sea was quite rough for the first fifteen knots, and the steering was not as straight as might have been, owing to the previous bad weather setting the buoys away. When this is taken into consideration, the *Maryland's* speed of 22.406 will place her at the top of the class.

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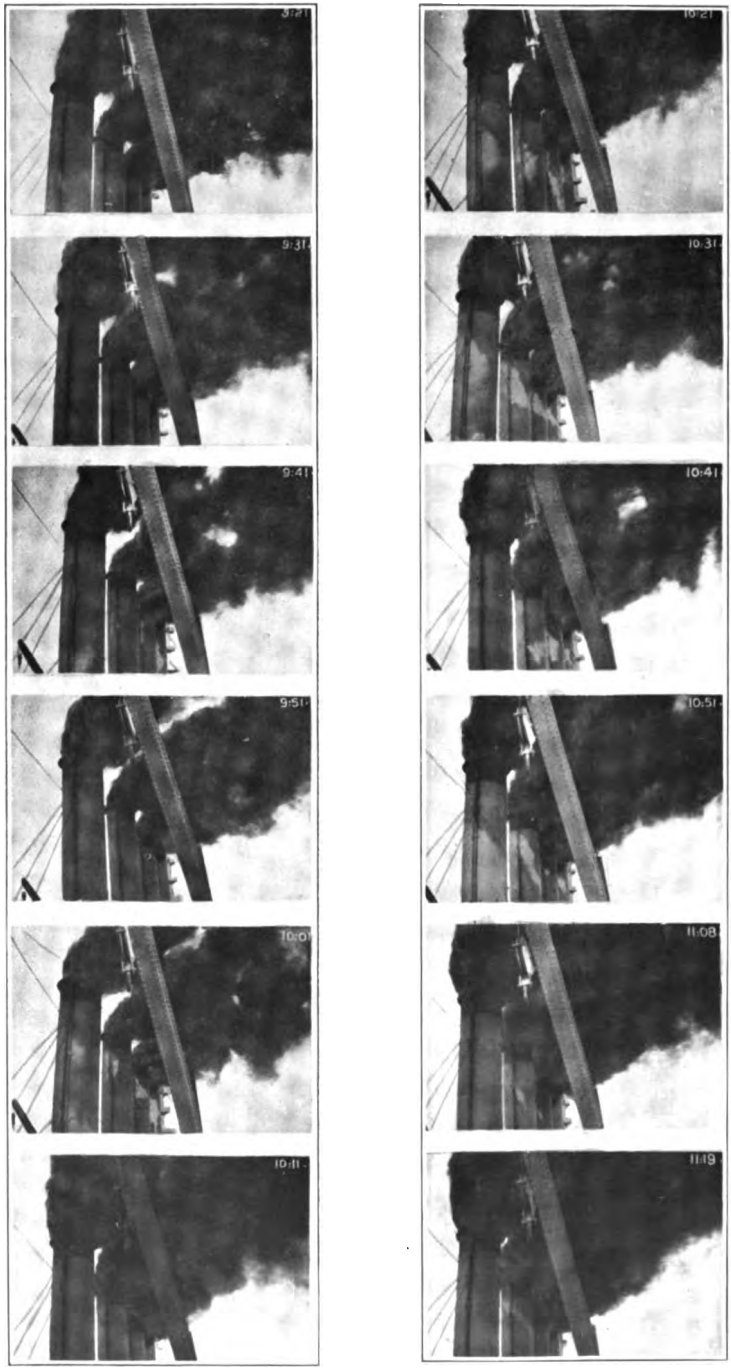

U. S. S. "MARYLAND."

Photographs of Smoke taken at ten-minute intervals during trial.
Run North.

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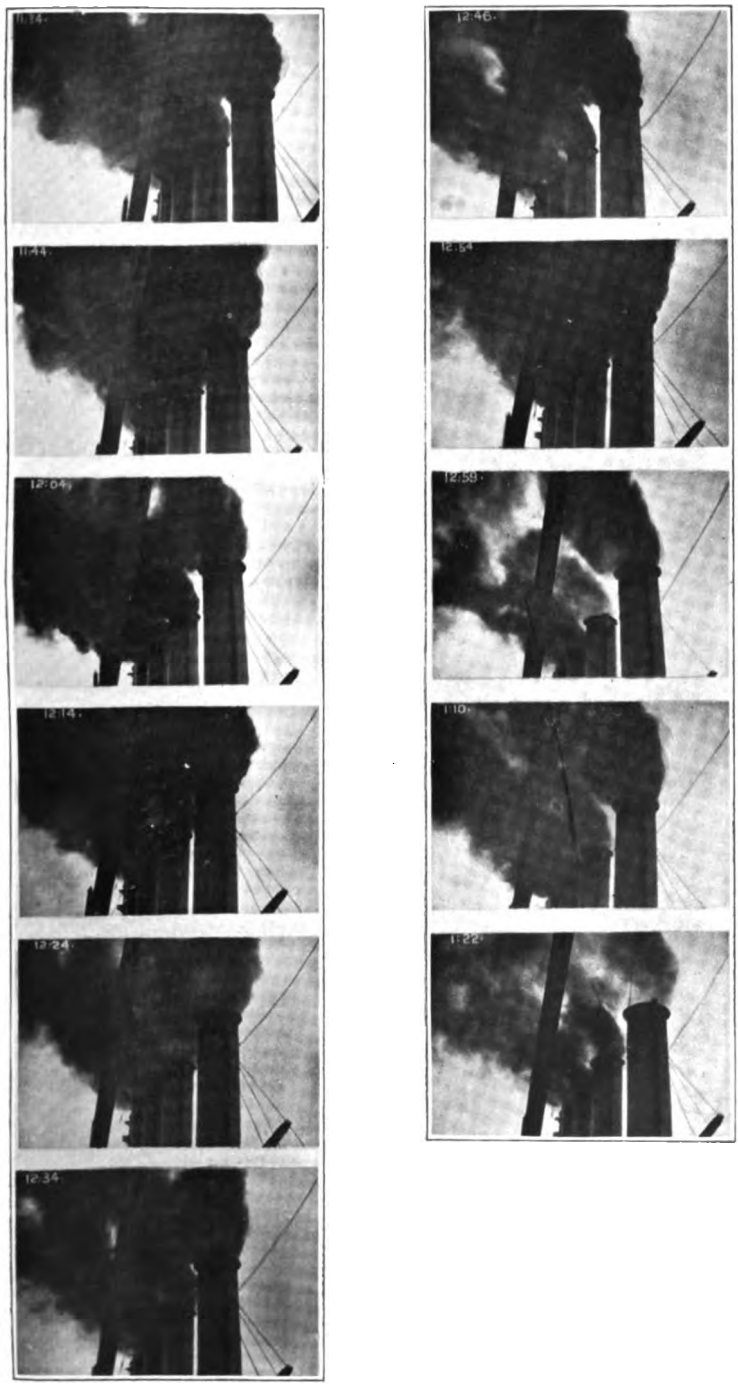

U. S. S. "MARYLAND."

Photographs of Smoke taken at ten-minute intervals during trial. Run South.

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REPORTS ON TESTS OF A CENTRIFUGAL PUMP AND A "SIROCCO" BLOWER DRIVEN BY DE LAVAL STEAM TURBINES.

1. The undersigned, having been appointed to make certain tests on a centrifugal pump as built by the De Laval Steam Turbine Company and a blower of the "Sirocco" type, both driven by De Laval Steam Turbines, together with steam consumption tests of the turbines, have the honor to submit the following reports:

2. The tests in question were conducted on November 30, December 1 and 2, 1904, in the testing room of the De Laval Steam Turbine Company's Works at Trenton, N. J., and were carried out with an object to determine the following items:

BLOWER.

1. The quantity of air delivered, corresponding pressures and revolutions per minute.
2. The air horsepower (A.H.P.).
3. The efficiency for normal and reduced output.

PUMP.

1. The quantity of water discharged, corresponding heads and revolutions per minute.
2. The water horsepower (W.H.P.).
3. The efficiency for normal and reduced output.

TURBINES.

1. Brake horsepower (B.H.P.).
2. Steam consumption per brake horsepower.
3. Their suitability in connection with pumps and blowers.

3. In explanation of remarks, possibly prompted by the comparatively narrow range within which these tests were carried out, it is necessary to mention that, owing to the time consumed

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in mounting the pump and blower separately, as well as disconnecting and fitting up for the brake tests and the demand upon the testing room for other tests of commercial importance, only such tests were made which would convey general information with regard to the adaptability for naval purposes of such pumps and blowers when driven by turbines.

DESCRIPTION OF APPARATUS.

4. Turbine-driven pumps and blowers being of comparatively recent application, none as yet having been installed in any ship of the Navy, the following short description of the machinery involved, and especially of the turbines, will be of interest.

5. The blower was of the "Sirocco" type, cased fan, of 20 inches diameter, double inlet, discharging on the side, mounted on a common bedplate with the turbine (Figs. 1 and 2). The turbine shaft was geared into one other shaft, to which the blower shaft was connected by a coupling driving the blower at one-tenth the number of revolutions of the turbine wheel.

6. The pump was of the centrifugal type, composed of two entirely independent pump casings with impellers, mounted with the turbine on a common bedplate (Fig. 3). The turbine shaft was geared into two other shafts, each one separately driving a pump. There was one suction and one discharge pipe for the two pumps, the discharge pipe of the first pump being connected directly to the suction pipe of the second pump, the size of the common pipe, 15 inches diameter, being that required by each pump separately. This arrangement of pumps is termed by the builders a centrifugal series pump, and is especially adapted where an extreme lift is required. It is, of course, readily understood that each pump may have an independent suction and discharge pipe, the system arranged in that way being called parallel-connected, discharging twice the quantity of water but against only one-half the head, the driving power in both cases being the same.

7. The De Laval steam turbine, Fig. 4, as is well known, is of pure impulse type, and consists of one single disc with one

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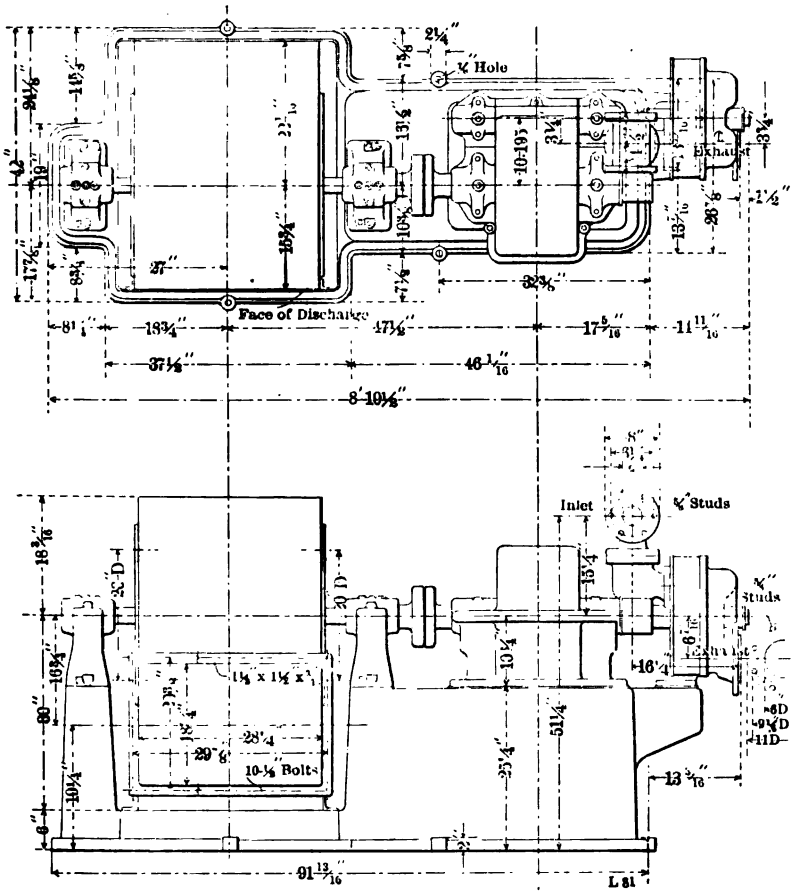

Fig. 1.

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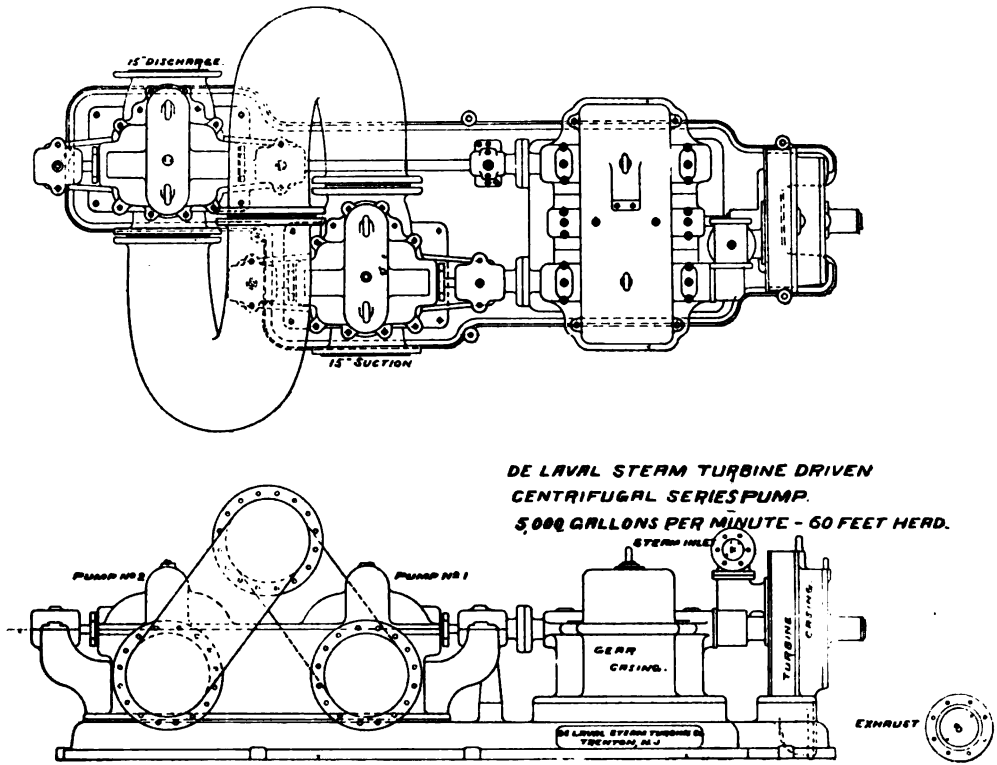

Fig. 3.

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row of vanes, against which steam impinges at a very high velocity as a result of expansion in diverging nozzles. These nozzles have their seating in a ring-shaped steam chamber, situated on the one side of the wheel with the exhaust chamber on the other, and are closely placed, occupying nearly the whole of the circumference. Each nozzle is controlled by a valve and hand-wheel independently.

8. The range of power is considerable, an increase or decrease in the amount of steam admitted to the wheel being governed by the number of nozzles open. The nozzles are proportioned with reference to the rotative speed of the turbine, and are made for either condensing or non-condensing conditions, the change being performed without a great deal of trouble.

9. The rotative speed of the wheel is kept constant for a variation in the load by a throttling device actuated by a centrifugal governor.

10. The gears of transmission consist of a spiral spur pinion made of high-grade, high-carbon nickel-steel, which engages either one or two spiral gears, as conditions may require. This depends largely on whether the power is wanted on one (as in the blower) or on two (as in the pump) shafts, but also on the size of the machine, the large machines invariably having double gears. The pinion is made solid with the turbine shaft, but the gears are built up of cast-steel pulleys with soft steel rings, of low carbon, shrunk on the rims.

11. The teeth are cut with perfect accuracy, resulting in almost noiseless motion, combined with great durability. Both pinion and gears are made in pairs, having the teeth inclined in opposite directions, thereby preventing distortion, due to the pressure of the gear itself, or to unequal thrust on the vanes, should such exist.

12. The main bearings are ring oiling, the oil being fed from a common reservoir placed at a convenient point and at some distance above the center of the shaft. The drip oil from the drain troughs is collected in a closed receptacle placed at the base and is under condenser pressure. No oil is used

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within the turbine itself, and, therefore, no oil exists in the exhaust steam.

13. Although rotating at an extremely high number of revolutions, there was but slight vibration, and only such noise as made by steam, water or air when flowing through pipes at a high velocity.

14. A feature of special interest are the coupling bolts and their connection to the couplings. The bolts are made of steel, but do not bear direct on the metal of the holes in the couplings, a rubber sleeve with a steel bushed hole of a slightly larger diameter than the bolt make up the combination. This is presumably done to reduce shock that may be effected by the rapid starting of the turbine wheel.

15. Superior workmanship, in connection with the use of high-grade materials, make this motor rank with the best, and, so far as known, its durability compares favorably with other motors, while its operation is marked by simplicity.

16. Its economic standard with regard to steam consumption may be judged from results obtained in the tests hereinafter referred to as well as numerous other tests made from time to time.

METHOD EMPLOYED TO MEASURE STEAM CONSUMPTION.

17. All De Laval steam turbines, unless specially ordered, are supplied with two independent sets of nozzles, one set being used when running condensing, the other when non-condensing, the number of nozzles in use being governed by the horsepower required.

18. Numerous experiments performed with De Laval nozzles having proved that only a certain amount of steam passes through the nozzle per unit of time for given conditions of pressure, vacuum and quality of steam, it was concluded that the apparatus used by the builders for the determination of steam consumption was sufficiently reliable. In one of its details this apparatus consisted of a pipe containing a nozzle of identical dimensions as those used in the turbine (condensing or non-condensing, as the case may be), the steam flow to

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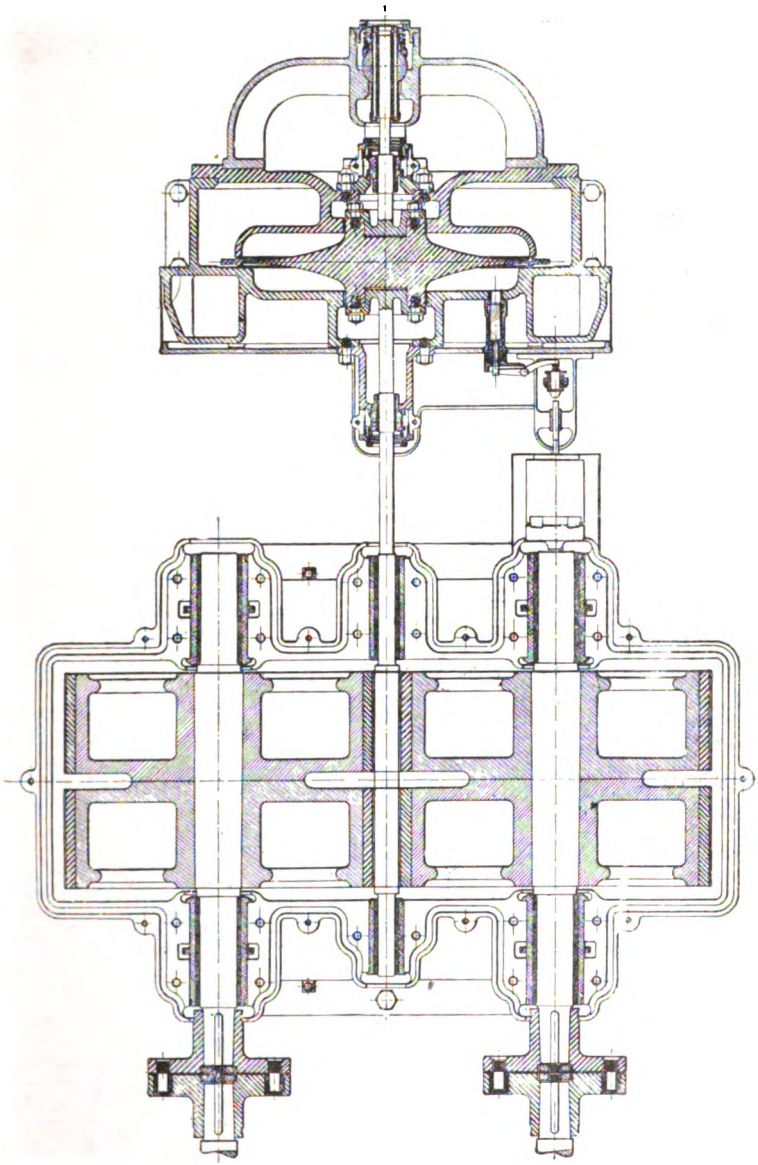

Fig. 4.

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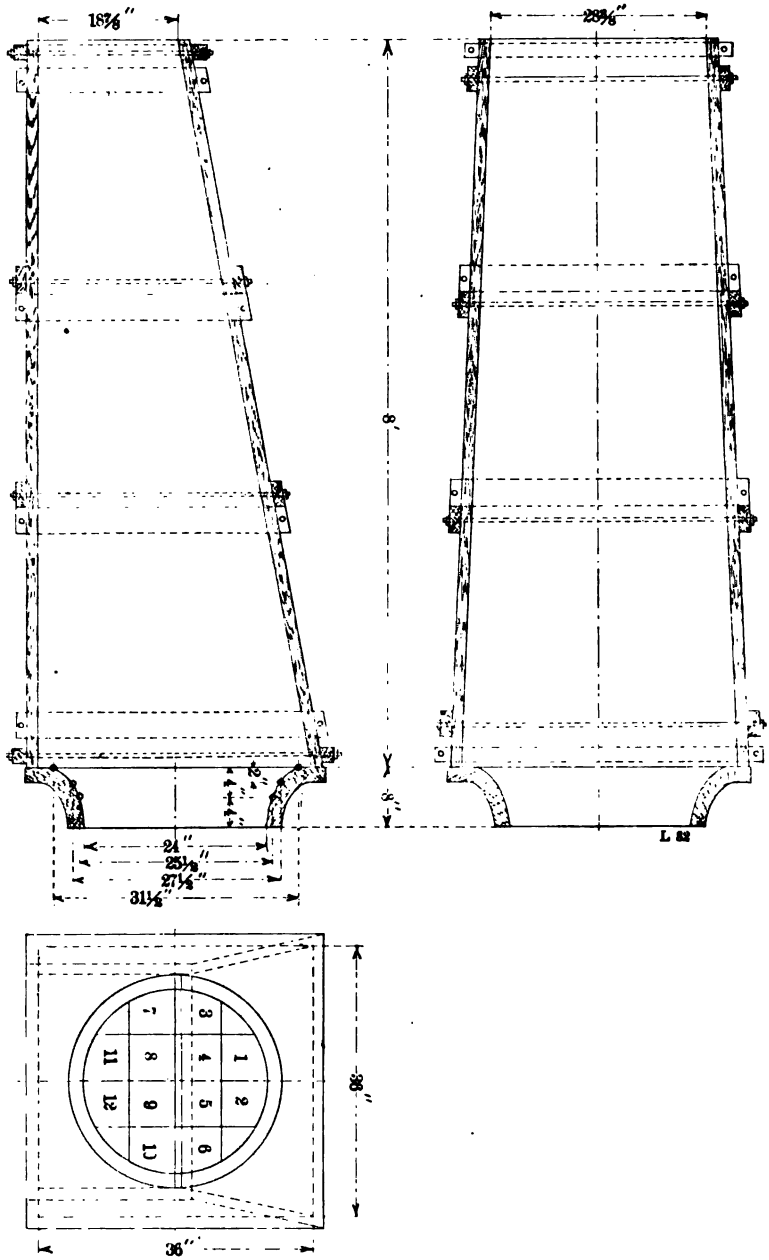

Fig. 6.

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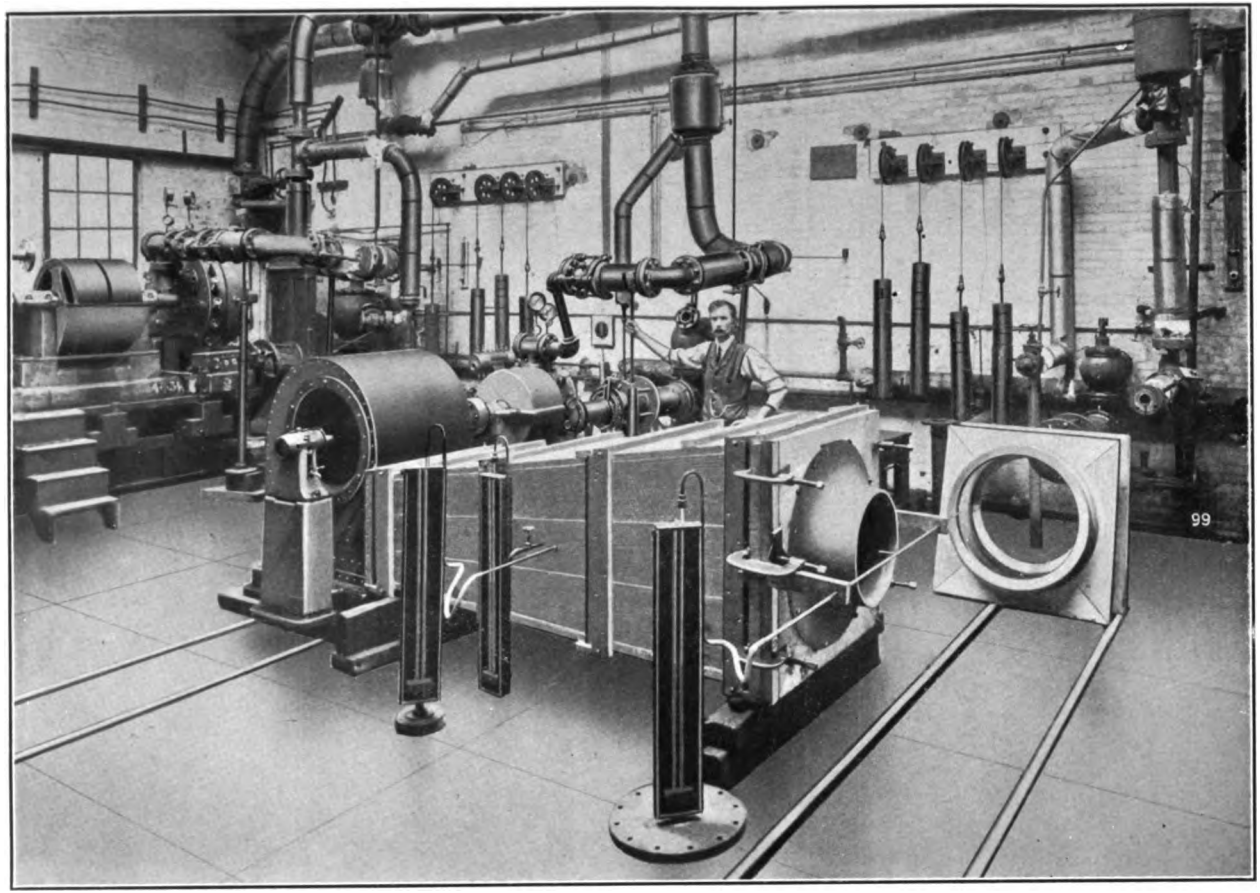

Fig. 2 and 5.

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it from the turbine casing being regulated by a three-way valve. The discharge end of the pipe was furnished with a perforated conical cap, and was located near the bottom of the condenser, consisting of a barrel containing water, placed on a scale. A steam gauge was placed between the nozzle and the barrel to ascertain with certainty that the pressure on the discharge side of the nozzle in no case exceeded 58 per cent. of that on the steam side, thus satisfying the condition for maximum discharge from a nozzle. In no case did this pressure rise to even nearly 58 per cent. The three-way cock was set immediately before each test to discharge all condensed steam in the pipe into the atmosphere, then quickly turned to admit the steam into the barrel. The time of admission, ranging between one and three minutes, was carefully noted, and the weight of the water in the barrel was taken before and after the discharge. The difference in these weights gave the total steam taken by the turbine by each nozzle for the time the nozzle was open. This is given in the tabulated data for each test separately, the total amount of steam used per hour being obtained by multiplying the quantity passed through test nozzle by the number of nozzles in operation into the ratio of sixty and the time it was open.

TEST OF SIROCCO BLOWER.

19. The blower in question was specially ordered for this test, and made to dimensions such that its output would conform to conditions required for the machinery installation of the three new scout ships, *Salem*, *Birmingham* and *Chester*. Under these requirements there must be delivered to each boiler room 56,000 cubic feet of air per minute furnished by one or two blower units, and a pressure of 5½ inches maintained.

20. The usual method of discharging the air from the blower into an air-tight receptacle was also employed in these tests, and is shown in Fig. 5.

21. At the outlet end of the plank box, shown in detail on

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Fig. 6, was fitted a converging nozzle of suitable proportions, with entrance line well rounded. The velocity head of the outflowing air was ascertained with a Pitot tube of $\frac{3}{32}$ inch diameter, the average for the whole nozzle being arrived at by measuring the velocity at different points of the orifice. These readings were obtained with accuracy by the following simple method:

22. The section at the outlet of the nozzle was divided in spaces, six inches square, by stretching fine wire across the orifice, and then, by recording the reading as obtained in the center of each square, the mean of all readings was taken, giving an average velocity head for the whole nozzle.

23. The pressure was ascertained by inserting in the box at each side tubes connected to syphon gauges. These tubes were furnished with bell-shaped mouth pieces, having their faces parallel to the flow of the air, and the readings were therefore not influenced by the velocity. Other Pitot-tube observations were made at about the middle of the box for three equidistant points. The temperature of the air in the box was taken near the compression gauge.

24. After completion of blower tests the blower was disconnected and brake tests of turbine performed, Fig. 7. This was done by connecting a Prony brake to the gear shaft and then by loading the lever until the steam gauge below the governor valve registered approximately the same pressure as during any one of the blower tests. The speed of the turbine as well as the number, etc., of nozzles open were adjusted to conform with conditions during the blower tests. Steam consumption was obtained in the manner previously described for the blower.

25. Tabulated results of various readings and observations will be found on Table I; results relating to steam consumption and other items for derivation of efficiency on Table II; calibrations of gauges on Table III.

Test "A" comprises blower performance at about 1,500 R.P.M., condensing (normal).

Test "B", same at about 1,000 R.P.M., condensing.

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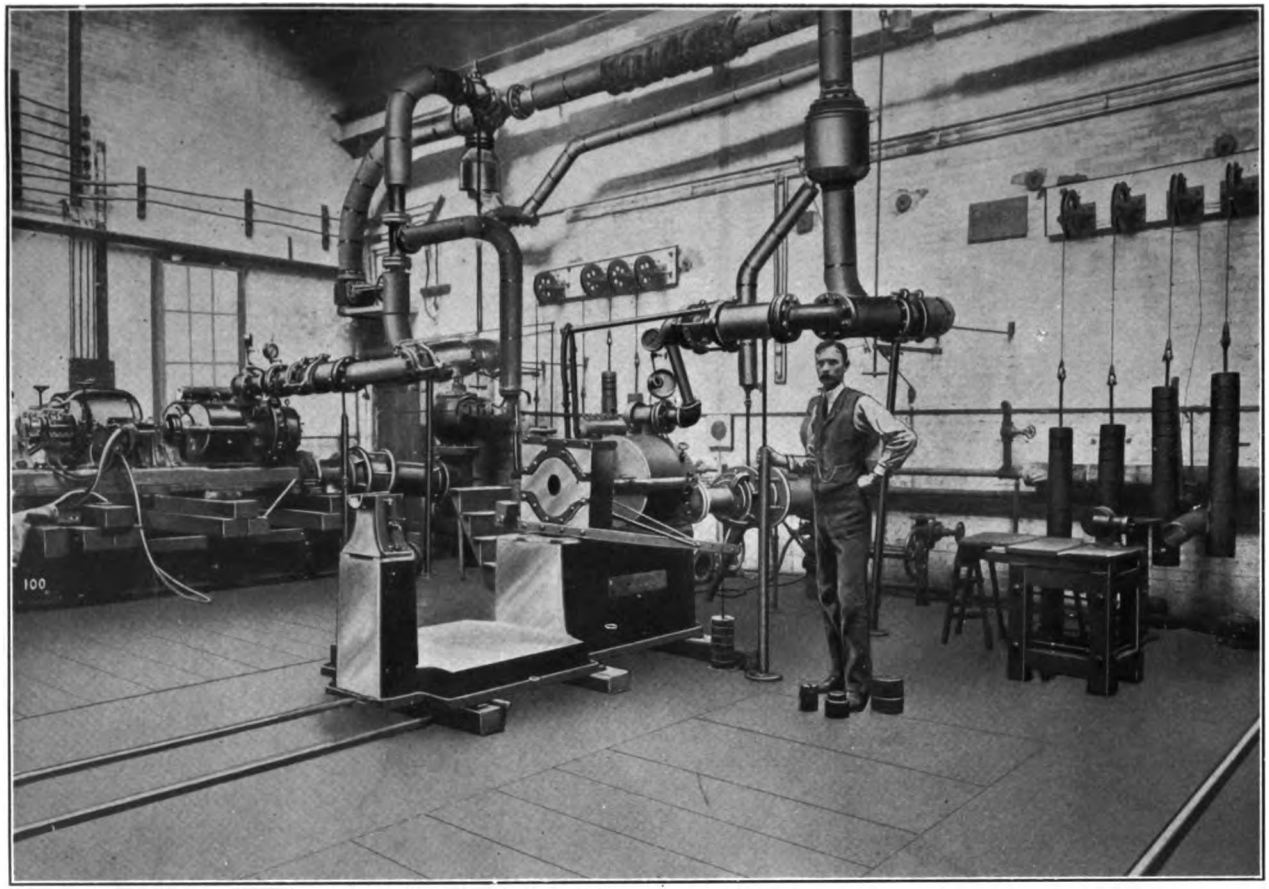

Fig. 7.

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TABLE I.

SIROCCO BLOWER TESTS.

Readings of Temperature, Barometer and Pressure in Air Box and at Pitot Gauge (inches of water).

Blower.	Test.	Number of test.	Time of test, year, 1904. Day	Hour.	Pitot tube in air box at three points. I*	II	III	Pitot tube at orifice of air box; readings taken at center in squares as designated in Fig. 6. 1	2	3	4	5	6	7	8	9	10	11	12	Compression in air box at three points. I*	II	III	Inlet to air box. Temp. air deg. C.	Press. ins. water.	Psychrometer. Deg. F. Wet	Dry	Barometer.
	A	1	November 30.	12·00	6 5/8	5 5/8	6 1/8	6 1/2	7	6 5/8	6 5/8	6 1/2	6 5/8	6 1/2	6 5/8	6 5/8	6 7/8	6 7/8	7	4 5/8	4 1/2	4 5/8	30	4 7/8	65	77	29.86
		2		12·10	6 3/4	5 7/8	6 1/8	6 1/4	6 7/8	6 1/4	6 1/2	6 5/8	6 1/2	6 5/8	6 5/8	6 5/8	6 3/4	6 7/8	7	4 5/8	4 1/2	4 5/8	30	5	66	77	
		3		12·20	6 5/8	5 3/4	6 1/8	6 1/4	7 1/8	6 5/8	6 5/8	6 5/8	6 5/8	6 1/2	6 1/2	6 5/8	7 1/4	7	7	4 9/16	4 1/2	4 5/8	30	5	66	76	
		4		12·30	6 5/8	5 1/2	6 1/8	6 1/4	7 1/8	6 1/4	6 1/2	6 1/2	6 1/2	6 1/2	6 1/4	6 5/8	7	6 3/4	7 1/8	4 5/8	4 1/2	4 5/8	30	4 7/8	66	79	
		5		12·40	6 3/4	5 5/8	6	6 5/8	7 1/4	6 1/4	6 1/2	6 3/4	6 5/8	6 5/8	6 5/8	6 1/2	7	6 3/4	6 7/8	4 5/8	4 1/2	4 5/8	30	4 7/8	65	78	
		6		12·50	6 7/8	5 1/2	6	6 1/2	7 1/8	6 5/8	6 1/2	6 1/2	6 1/2	6 1/2	6 5/8	6 5/8	6 7/8	6 7/8	7 1/8	4 5/8	4 1/2	4 1/2	30	4 7/8	65	78	
		7		1·00	6 5/8	5 3/4	6	6 5/8	7	6 1/4	6 1/2	6 1/2	6 5/8	6 1/2	6 1/2	6 1/2	6 3/4	7	6 1/2	4 9/16	4 5/8	4 5/8	30	5	66	79 1/2	
	Averages				6.17			6.64												4.6			.....		65.6	78.3	.....
	B	8	Nov. 30.	2·00	3 1/4	3 1/4	2 7/8	3	3 5/8	2 7/8	3	3 1/16	3 1/8	2 15/16	3	3	3 3/16	3 1/8	3 3/16	2 1/8	2 1/8	2 3/16	27 1/2	2 5/8	63 1/2	75 1/2	29.86
		9		2·10	3 1/4	3 1/4	2 15/16	2 7/8	3 1/4	2 7/8	3 1/16	3 1/8	3 1/16	3 1/16	3	3 1/8	3 5/16	3 3/16	3 1/4	2 1/8	2 1/8	2 1/8	27 1/2	2 5/8	61 1/2	75	
		10		2·20	3 3/16	3 3/16	2 15/16	3 1/8	3 1/4	2 15/16	3 1/8	3 1/16	3	3	3 1/16	3 1/16	3 1/4	3 1/8	3 1/8	2 1/8	2 1/8	2 1/8	27	2 7/16	62	75	
		11		2·30	3 3/16	3 3/16	2 15/16	2 7/8	3 1/8	2 7/8	3	3 1/16	3	2 7/8	3	3	3 1/4	3 1/8	3 1/4	2 3/16	2 1/16	2 1/16	25 1/2	2 5/16	59	71	
	Averages				3.063			3.075												2.125			.....		61.5	74.1	.....

* Center.

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TABLE II.

SIROCCO BLOWER TESTS.

Average Readings of Numbered Tests in Each Group.

	Condition.	Test.	Number of test.	Time of test, November 30, 1904.	Steam pressure in pounds at governor valve.		Temp. of steam, deg. C.	Vacuum, ins. mercury.	Revolutions per minute.	Number nozzles open.	Test noz. open in minutes.	Net steam condensed in pounds.	Total steam used in pounds per hour.	Air horsepower, A.H.P.	Brake horsepower, B.H.P.	Efficiency, per cent. A.H.P. ÷ B.H.P.	Steam in lbs. per hour, per A.H.P.	Steam in lbs. per hour, per B.H.P.	Air discharged, in cubic feet, per minute.	Load on brake lever, lbs.
				Hour.	*Above.*	*Below.*														
Blower.	Con.*	A	1, 2	12·00–12·20	194.5	164	196.3	$25\frac{3}{4}$	1507.5	7	2	7	...	...	...	...	...	...	...	...
			3-5	12·20–12·50	193.3	164	197.6	$25\frac{5}{8}$	1501.4	7	2	7	...	...	...	...	...	...	...	...
			6, 7	12·50– 1·10	193.5	163.5	197	$25\frac{7}{8}$	1500	7	2	6.6	...	...	...	...	...	...	...	...
	Aver.		...	...	193.9	163.7	197	25.84	1502.8	7	2	6.866	1,441.86	34.45	63.54	54.2	42.09	...	33,170	...
	Con.*	B	8, 9	2·00– 2·20	170	167.75	188	$26\frac{5}{16}$	1041.5	3	2	7.1	...	...	...	...	...	...	...	...
			10, 11	2·20– 2·40	166	164	187.5	$26\frac{7}{16}$	1022.5	3	2	6.9	...	...	...	...	...	...	...	...
	Aver.		...	...	168	165.8	187.75	$26\frac{3}{8}$	1031.5	3	2	7	630	10.81	22.98	47.05	58.26	...	22,230	...
Turbine.	Con.*	C	12–15	3·55– 4·20	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
	Aver.		...	...	170	163.5	187.2	26.38	1048.7	3	2	6.6	594	...	21.67	...	...	27.41	...	31
	Con.*	D	16–19	4·45– 5·15	...	...	...	...	...	...	...	...	...	...	...	...	..	...	...	...
	Aver.		...	...	195	160.25	195.5	25.48	1532.2	7	2	7	1,470	...	64.77	...	...	22.69	...	63.43
			1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18

*Condensing.

NOTE.—In tests, groups A and D correspond, and B and C correspond.

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Test "C", turbine performance at about 1,500 R.P.M., blower disconnected, condensing.

Test "D", same at about 1,000 R.P.M., condensing.

TABLE III.

CALIBRATION OF GAUGES USED DURING TEST OF BLOWER.

Standard gauge.	Gauge below governor valve.	Gauge above governor valve.
150	150	155
155	155	160
160	160	165
165	165	170
170	170	175
175	175.5	180
180	180.5	185
185	186.5	190
190	191	195
195	...	200
200	...	205

Tested December 1, 1904.

CALCULATION OF BLOWER RESULTS.

26. *Air Delivery.*—In order to obtain accurate figures for the density of the air and its dependent velocity head, the usual observations of temperature, atmospheric pressure and also moisture were made, the results being recorded in Table I. The amount of moisture contained in the air influences final results but slightly, except during extraordinay atmospheric conditions, but is noted so as to obtain more absolute comparison with other tests made on blowers.

27. The following formulae will give the density with relation to foregoing items:

$$W = \frac{0.0807\,(p - 0.379\,h \times e)}{(1 + \propto t)\,29.921},$$

where

0.0807 = the weight of 1 cubic foot of dry air at 32 degrees Fahrenheit, and at a normal atmospheric pressure of 29.921 inches.

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p = atmospheric pressure;
h = relative humidity;
e = vapor pressure;

$1 + \propto t$ = the volumetric ratio of dry air at temperature 32 degrees Fahrenheit and at $(32 + t)$ degrees Fahrenheit;
t = temperature of dry air above 32 degrees;
.379 = the percentage of vapor contained in one pound of saturated air at 32 degrees.

Then for average results in Test "A", the following is obtained:

$$h = .55;\ e = 1.015;\ t = 45.5 \text{ degrees F.};\ p = 29.86;$$
$$1 + \propto t = 1.093;$$
$$W = \frac{0.0807\ (29.86 - 1.015 \times .55 \times .379)}{1.093 \times 29.921} = .07298.$$

The velocity of discharge is $V = \sqrt{2g\ H}$ feet per second, H being the height of an air column which balances the mean height of water columns recorded in Pitot tube readings, and thus:

$$H = \frac{62.24 \times 6.64}{12 \times 0.07298} = 471.7 \text{ feet, and}$$
$$V = \sqrt{2 \times 32.16 \times 471.7} = 174.2 \text{ feet per second.}$$

The amount of air discharged per second is $Q = C \times A \times V$, in which equation

A = area of nozzle in square feet = 3.15;
C = a constant allowing for the contraction of air current in the nozzle. This constant, owing to size and shape of nozzle (See Fig. 5), may, without an appreciable error, be made equal to 1, and
$Q = 3.15 \times 174.2 = 549.5$ cubic feet per second
$= 60 \times 549.5 = 33{,}170$ cubic feet per minute.

Brake horsepower: For Test "D", corresponding to Test "A", of blower—Length of lever, 3 feet 6 inches; load on lever, 63.43 pounds; number revolutions of pulley = 1,532.2.

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$$H.P. = \frac{6.28 \times 3.5 \times 1{,}532.2 \times 63.43}{33{,}000} = 64.77, \text{ Test "D", normal.}$$

$$= 21.67, \text{ Test "C", reduced.}$$

Steam per brake horsepower: Total steam used per hour divided by brake horsepower $= \frac{1{,}470}{64.77} = 22.69$ pounds per hour.

It will be noticed that the turbine ran about 30 revolutions faster during Test "D" (brake) than during Test "A" (blower), and the brake horsepower of the blower, therefore, deduced from total steam consumed by blower and steam per brake horsepower will be:

$\frac{1{,}441.86}{22.69} = 63.54$ and $\frac{630}{27.41} = 22.98$ respectively for "*A*" and "*B*".

Air horsepower: The energy in foot pounds developed by the air discharged per second from the nozzle moving at a velocity V must equal the work expended upon it, thus:

$$A.H.P. \times 550 = 0.07298 \times 549.5 \times \frac{174.2^2}{2g} \text{ and}$$

$$A.H.P. = 34.45 \text{ for conditions in Test "A" and}$$
$$= 10.81 \text{ for conditions in Test "B".}$$

Efficiency of blower: Dividing air horsepower with brake horsepower will give the efficiency;

$$\text{Then, } E = \frac{34.45}{63.54} = 54.2 \text{ per cent. at normal output,}$$

$$\text{Then, } E = \frac{10.81}{22.98} = 47.05 \text{ per cent. at reduced output.}$$

Steam per air horsepower: For normal conditions, 42.09 pounds per hour. For reduced conditions, 58.28 pounds per hour.

A synopsis of data with reference to these tests will be found on Table II.

29. In comparison with fans of the Sturtevant type, installed on the U. S. S. *Minneapolis*, and driven by a set of two-cylinder reciprocating, simple engines, the following is submitted. The figures are derived from steam-consumption tests made by W.

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W. White, U. S. N. (See Volume X, JOURNAL OF AMERICAN SOCIETY OF NAVAL ENGINEERS.)

ENGINE BLOWER. *Sturtevant, Test.*		TURBINE BLOWER. *Sirocco, Test.*	
Air pressure, inches	.7	Air pressure, inches	2.1
Steam pressure, pounds	120	Steam pressure, pounds	164
Vacuum, inches	15¼	Vacuum, inches	26.34
I.H.P.	16.3	I.H.P.	...
B.H.P. (based on combined mechanical efficiency of engine and blower equal to .8)	13.04	B.H.P.	22.98
A.H.P. (calculated)	7.75	A.H.P. (as tested)	10.81
Efficiency of blower, per cent.	59.45	Efficiency of blower	47.05
Steam per A.H.P., pounds	162	Steam per A.H.P.	58.26
B.H.P.	96.82	B.H.P.	27.41

NOTE.—Figures above refer to two-thirds capacity of normal for turbine blower and thus, far below those given for normal output.

30. The air horsepower of engine blower is figured by the usual formula, (no tests of air discharge having been performed when tests for steam consumption were carried out); while the figures given for the turbine blower are taken and put down from the tests. The result of the comparison shows however, that the steam used per Brake horsepower in the two different systems is much in favor of the turbine, and would be under non-condensing conditions with the turbine, even though it be granted that the steam consumption then would be twice as great, which, as will be seen later in the pump tests, is not quite the condition.

31. The efficiency of the Sirocco fan is apparently not as good as the ordinary type; but as this fan adapts itself better for a turbine drive with its high number of revolutions, it seems justifiable to recommend its adoption for certain purposes.

32. Furthermore, values of efficiency probably will be somewhat modified by variations in conditions, and final conclusions should not be drawn from the few tests made; but additional tests should be performed with Sirocco fans when run with

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turbines, to enable the determination of conditions that will give maximum efficiency as well as the limits within which said blowers may be used to advantage.

33. Attention is drawn to the following points:

1. In comparison with blowers shown in official plans of the new scout cruisers, the "Sirocco" turbine blower takes up less room, as per following table:

Sturtevant-Engine.		*Sirocco-Turbine.*	
Length, inches	96	Length, inches	96
Breadth, inches	70	Breadth, inches	42
Height, inches	84	Height, inches	54
R.P.M	418	R.P.M	1,500
Diameter of fan, inches	66	Diameter of fan, inches	20
Width of fan (tip), inches	20	Width of fan (tip), inches	24
Weight, pounds	2,650	Weight, pounds	4,360

The greater weight of the turbine blower is readily accounted for by the extremely heavy bedplate upon which it was mounted. The bedplate, however, may be considerably reduced in weight when designed to conform to installations on board ship.

2. A turbine-driven blower works steadier and with less noise; it does not require any special care in adjustments when in operation, and is probably less liable to break down at speeds necessary to give the high air pressures required in special cases.

3. It is more economical, both in steam and in oil required.

TEST OF CENTRIFUGAL PUMP.

34. The pump in question was built for Messrs. Jones & Laughlin, of Pittsburg, Pa., and was intended for service in connection with the local fire system within their works. It was mounted, together with the turbine, in a convenient position to draw from and discharge into the storage tanks built under the floor of the testing room. The discharge pipe was furnished with a converging nozzle of such shape that the loss due to contraction at the outlet was considered negligibly small. (See Fig. 8.)

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35. The velocity head was recorded in accordance with readings obtained with a Pitot tube in connection with a vertical glass gauge fastened to the wall and graduated in inches for a height of about 28 feet. The output of the pump was determined by this method alone, weir tests being impracticable, owing to the limited capacity of the tanks.

36. The rated capacity of the pump was about 10,000 gallons per minute with a thirty-foot total lift, but, owing to the limited capacity of the tanks and to difficulties then arising when handling such large amounts of water, it was arranged to discharge one-half this quantity, but against a sixty-foot head, the average water horsepower of the latter being equivalent to the first, other conditions being similar.

37. Four tests of the pump and four tests with the turbine disconnected from the pump were performed. In each series two tests comprised condensing conditions, while the other were non-condensing. Tabulated average results of Pitot tube readings relating to suction and discharge heads, together with other results, will be found in Table IV.

Test "E" constitutes pump performance at about 1,200 R.P.M., condensing (normal).

Test "F," same at about 975 R.P.M., condensing.

Test "G," same at about 975 R.P.M., non-condensing.

Test "H," same at about 1,200 R.P.M., non-condensing.

Test "K" constitutes turbine performance at about 1,200 R.P.M., non-condensing.

Test "L," same at about 975 R.P.M., non-condensing.

Test "M," same at about 975 R.P.M., condensing.

Test "N," same at about 1,200 R.P.M., condensing.

At the finish of each complete series of tests the gauges were removed and calibrations carefully made. The results will be found in Table V.

While performing the brake tests the pump was disconnected from the turbine and a Prony brake connected to each gear shaft. Adjustments were made to bring about uniformity in the conditions for corresponding tests.

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TABLE IV.

CENTRIFUGAL PUMP TESTS.

Average Reading of Numbered Tests in Each Group.

	Condition.	Test.	Number of test.	Time of test, year 1904.			Steam pressure in pounds at governor valve.		Temp. of steam, degrees C.	Vacuum, inches mercury.	Revolutions per minute.	Number of nozzles open.	*Test nozzle open, minutes.	*Net steam condensed, lbs.	Total steam, pounds per hour.	Water horsepower, W.H.P.	Brake horsepower, B.H.P.	Efficiency, per cent. W.H.P. ÷ B.H.P.	Pounds steam per hour per W.H.P.	Pounds steam per hour per B.H.P.	Water discharge, gallons per minute.	Vacuum in suction pipe at pump, inches mercury.	Pressure in discharge pipe at pump, lbs. per square inch.	Total lift in feet.	Pressure in Pitot tube.	Load in pounds on brake lever.
				D'y.	Hour.		Above.	Below.																	ft. in.	
Centrifugal pump.	Condensing.	E	20, 21, 22, 23	December 1.	11·25 to 11·55		195	172	192.7	26.14	1,215	7	2	9	1,890	76.7	...	72.1	24 64	...	5,110	7.562	21.6	59.4	18–10	...
		F	24, 25, 26, 27		12·35 to 1 05		185	184	187	26.78	974	4	2	9.3	1,116	42	...	62 3	26.57	...	2,640	5.187	24.35	62.98	8–11¼	...
	Non-condensing	G	28, 29 30		2 10 to 2·30		160	158 4	182	0	996	3	1.5	20 2	2,430	43.3	...	61.7	56.12	...	2,660	5.075	25	64.46	9– ¾	...
		H	31, 32, 33		2 55 to 3·15		192.6	180 5	191.6	0	1,213	4	1	14 5	3,480	77.1	...	71.6	45.15	...	5,120	7.542	21.73	59.66	18–10¾	...
Turbine.	Non-condensing.	K	34, 35, 36	December 2.	10·15 to 10 35		194	178.6	191.1	0	1,212	4	1	14 3	3.435	...	108.2	...	...	31.7	...	...	...	...	...	104.53
		L	37, 38, 39		11·05 to 11·25		165	168.5	188	0	968	3	1.5	20	2,400	...	70.1	...	...	34.2	...	...	...	...	...	84.5
	Condensing.	M	40, 41, 42		12·45 to 12·55		186	185	190	27.125	967.5	4	2	10	1,200	...	66 3	...	...	18 1	...	...	...	...	...	80
		N	43, 44, 45		1·00 to 1·10		194.5	169.75	190	26.187	1,209	7	2	8 6	1,805	...	106.3	...	...	16 95	...	...	...	...	...	102 55
		1	2	3	4		5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24

NOTE.—Tests E and N correspond, F and M correspond, G and L correspond, and H and K correspond. Size of steam nozzles, 4.7 mm. condensing; 8.7 mm. non-condensing. Size of water nozzles, No. 32, or 7 947 inches for E and H; No. 24, or 6.882 inches for F and G. Temperature of air, 20 degrees C; of water, 15 degrees C.
*Only one observation in each group.

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TABLE V.

CALIBRATION OF GAUGES USED DURING TEST OF PUMP.

Standard gauge.	Gauge No. below governor valve.	Gauge No. above governor valve.	Gauge at delivery of pump.
15	...	...	14
20	...	...	19.1
25	...	...	24.2
30	...	...	29.5
165	164.5	170	...
170	170	175	...
175	175	180	...
180	180.5	185	...
185	186	190	...
190	191	195	...
195	196.5	200	...
200	201.5	204	...
...	...	209	...

Tested December 2, 1904.

CALCULATION OF PUMP RESULTS.

38. *Water discharge:* Weir tests, as already mentioned, not being practicable, it was necessary to make calculations from data obtained on the "Pitot tube basis." Experiments have shown (some of recent date by Professors Denton and Kent,) that the quantity discharged may accurately be ascertained from the formula:

$$Q = C \times \text{area of nozzle} \times \sqrt{2gH}$$

with a value of C varying between .953 and .977 and may, with reliability, be assumed to equal at least .95. For average readings in Test "E", $H = 18$ feet 10 inches, the height of water column in Pitot tube, and

$$Q = .95 \times \frac{49.54}{144} \sqrt{64.32 \times 18.833} = 11.39 \text{ cubic feet per second}$$
$$= 5{,}110 \text{ gallons per minute.}$$

39. *Brake horsepower:* From data corresponding to conditions in Test "E".

Length of lever, feet and inches	4- 6
Load on lever, pounds	102.55
Number revolutions of pulley	1,212

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$$B.H.P. = \frac{6.28 \times 4.5 \times 1{,}212 \times 104.55}{33{,}000} = 106.3.$$

40. *Steam per brake horsepower:* For Test "E"

$$= \frac{1{,}805}{106.3} = 16.95 \text{ pounds per hour.}$$

40. *Water horsepower:* From Table IV, total lift, obtained by adding suction and discharge heads, equals 59.4 feet.

$$W.H.P. = \frac{5{,}110 \times 59.4 \times 8.376}{33{,}000} = 76.7.$$

41. *Efficiency of pump:* Water horsepower divided by brake horsepower gives efficiency, and thus:

$$E = \frac{76.7}{106.3} = 72.1 \text{ per cent. at normal output.}$$

42. *Steam per water horsepower:* For Test "E".

$$\frac{\text{Total steam}}{W.H.P.} = \frac{1{,}890}{76.7} = 24.64 \text{ pounds per hour.}$$

A synopsis of results made up from average readings will be found on table IV.

43. The efficiency, although good, is not as high as has been obtained at other times with this type of pump. In the results obtained with centrifugal pumps built for the U. S. A. C. *West Virginia*, and tested at the works of the Newport News Shipbuilding Company, of Newport News, Va., the following comparative efficiency is submitted:

Engine pump, 61.6 per cent.
Turbine pump, 72.1 per cent.

The efficiency of the engine pump was figured on the basis of brake horsepower, which was obtained by multiplying the I.H.P. with an assumed combined mechanical efficiency of .8 for engine and pump.

44. The reason of a higher efficiency with turbine pumps will no doubt be found due to the considerably lessened frictional resistances in the small impellers used in those pumps. Frictional resistances of revolving discs, as have been found

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experimentally, increase in the fifth power of the radius, while only in the third power of the revolutions.

45. With reference to steam consumption of the turbine pump when placed in comparison with an engine-driven pump, the following data are submitted. These data are taken from the same source as those previously referred to in the Sirocco blower tests.

Engine pump.		*Turbine pump.*	
Type of engine:		Type of turbine:	
Simple, two-cylinder, slide valve.		De Laval condensing.	
Steam pressure, pounds	154	Steam pressure, pounds	185
Vacuum, inches	13	Vacuum, inches	27.1
I.H.P.	18.87	I.H.P.	...
B.H.P. (.8 I.H.P.)	15.1	B.H.P.	66.3
Steam per B.H.P., pounds	36.5	Steam per B.H.P., pounds	18.1
R.P.M	172	R.P.M	968

NOTE.—Above conditions given for turbine pump correspond to one-half the normal capacity.

46. The following comparative figures give space and weight of the ordinary type of centrifugal pumps as shown in the official plans of the new scout cruisers and the turbine-driven pump described in tests. Capacity is the same in both cases, or 10,000 gallons per minute.

Engine pump.		*Turbine pump.*	
Length, feet and inches	8–6	Length, feet	13
Width, feet and inches	4–6	Width, feet	5
Height, feet and inches	6–6	Height, feet and inches	3–3
Weight, pounds	9,520	Weight, pounds	12,900

47. The following may be inferred from the foregoing tests and figures:

1. For small heads, such as is common on board ship for bilge or circulating purposes, the ordinary single-stage pump, although more wasteful of steam, seems better adapted than the series or multiple-stage pump, the former weighing less and taking up less room. For fire service or distiller circulating purposes the turbine pump may be used with decided advantage.

2. The turbine pump excels in efficiency the ordinary centrifugal pump, as shown by the foregoing tests.

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3. In its use of steam the turbine-driven pump is superior to the ordinary type, depending principally on the greater economy of the turbine, and in a lesser degree on the higher pump efficiency.

CONCLUSIONS.

a. Steam turbines inherently possess advantages over the reciprocating engine for driving centrifugal machines, and will, in our opinion, in many instances, replace the engine in driving both centrifugal pumps and blowers.

b. The De Laval steam turbine, being specially adapted for powers below 300 H.P., and, unlike the engine, possesses an acceptable degree of efficiency when built in small units, will be superior, especially when placed in positions where temperature and cramped facilities render the engine undesirable on account of its need of constant attention.

c. Whether reduction in weight or saving of space actually will be brought about by the adoption of these motors cannot definitely be determined before comparative trials have been made and a broader field opened to the turbine makers, enabling them to specialize their machines to suit particular naval requirements.

d. It is recommended that an installation of turbine-driven centrifugal pumps and forced-draft blowers be made for at least one of the scout cruisers to be contracted for by the Navy Department in the near future.

Very respectfully,

A. B. CANAGA,
Commander, U. S. N.
H. C. DINGER,
Lieutenant, U. S. N.
ERNEST N. JANSON,
Of the Bureau of Steam Engineering,
Navy Department.

THE CHIEF OF THE
BUREAU OF STEAM ENGINEERING,
Navy Department.

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THE U. S. S. *CHATTANOOGA.*

GENERAL DESCRIPTION—STANDARDIZATION AND OFFICIAL TRIALS.

BY LIEUTENANT COMMANDER W. W. WHITE, U. S. NAVY, MEMBER.

The sheathed protected cruiser *Chattanooga* (Cruiser No. 16) is one of six (designated as the *Denver* class) contracted for by Lewis Nixon (Crescent Shipyard) of Elizabethport, N. J., on December 14, 1899, to be finished in thirty months. In event of delay in completion a penalty of $300 was to be exacted for each day (except Sundays) for the first month, after the expiration of the contract time, $400 for each day of the second month, and thereafter at the rate of $600 per day. * Machinery weights were limited to 427 tons, with a penalty of $500 per ton for excess weight, and an additional fine of $10,000 in case the total weight as specified was exceeded by 5 per cent.

SCHEDULE OF WEIGHTS (DESIGNED).

Hull and fittings, not including protective-deck armor, tons	1,706
Protective-deck armor, 2-inch, tons	62
Cellulose, tons	24
Armament and ammunition, tons	213
Equipment, stores and outfit, tons	287
Propelling machinery, with water, tons	441
Coal, tons	467
Total	3,200

The speed to be developed and maintained successfully for four consecutive hours was to be not less than an average of 16½ knots, the air pressure in the firerooms being limited to 1 inch of water. In case the speed fell below 16½ knots on trial, but above 15½ knots, the vessel was to be conditionally accepted, subject to a deduction from the contract price of the

* Includes engines, boilers and appurtenances, all fixtures in engine and firerooms, smoke pipes, distilling apparatus, stores, spare parts, heating apparatus, tools in workshop, water in boilers, condensers, pumps, pipes and stern tubes (but not including reserve feed water, capstan, windlass, steering gear or winches).

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vessel on account of such failure at the rate of $25,000 a quarter of a knot for speed between 16½ and 16, and $50,000 a quarter knot between 16 and 15½ knots. If the vessel failed to exhibit an average speed of 15½ knots, her acceptance, at a reduced price to be agreed upon, was optional.

The keel was laid on the 29th of March, 1900, and after many delays the ship was launched on March 7, 1903, her contract date of completion having been June 14, 1902. When the shipbuilding trust, of which the Crescent Shipyard was a member, collapsed, the vessel was taken possession of by the Government, under the twelfth clause of the contract, which provided that in case of failure or omission of the contractor, at any stage of the work, to make satisfactory progress towards completion of the vessel the contract was to be forfeited. The vessel was removed to the navy yard, New York, about the middle of October, 1903, to be completed by the Government.

A Board of five naval officers was appointed to make a complete inventory of all work done or commenced upon the vessel, and all material furnished or on hand for the purposes of construction. This Board, under date of October 7, 1903, reported in part as follows:

"The original contract price of the *Chattanooga* was one million thirty-nine thousand nine hundred sixty-six dollars ($1,039,966). This is apportioned between hull (including equipment and ordnance) and machinery as follows:

Hull, ...	$639,966.00
Machinery, ...	400,000.00
	1,039,966.00

"Changes in the hull have increased the portion allotted to it by the sum of $3,806.52, making the revised hull allotment $643,772.52.

"Changes in the machinery have decreased the portion allotted to it by the sum of $834, making the revised machinery allotment $399,166.

"The revised hull and machinery allotments give a total, for basis of valuation, of $1,042,938.52.

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Amount earned on hull, including extras, . .	$449,529.30
Amount earned on machinery, including extras,	320,382.00
Total amount earned, ...	769,911.30
Cash paid on hull, ...	422,891.64
Cash paid for extras, ...	1,764.00
Cash paid on machinery, ...	325,883.88
Total cash paid contractor, ...	750,539.52
Total amount earned (estimated), ...	769,911.30
Total amount paid, ...	750,539.52
Balance due contractor, ...	$19,371.78

"The amount of seven hundred sixty-nine thousand nine hundred eleven dollars and thirty cents ($769,911.30) the Board declares to be a fair market value of the work and material belonging to the *Chattanooga*, including a reasonable margin of profit upon so much of the work as has been satisfactorily performed at the time of the forfeiture of the contract, the cost price plus extras being taken as the fair market value of the completed vessel."

It will be seen from the above that at the time of removal of the vessel to the New York Navy Yard there remained available, in accordance with the contract price, the sum of $273,027.22. Of this amount $194,243.22 was for finishing the hull and $78,784 the machinery.

As complete descriptions of this class of ships have appeared at various times in the JOURNAL, it will be only necessary to repeat here the principal dimensions. For more extended data, the reader is referred to Vol. XV, page 1110 (Trial of U. S. S. *Cleveland*), Vol. XVI, page 1 (U. S. S. *Tacoma*), page 67 (U. S. S. *Denver*), and page 227 (U. S. S. *Des Moines*).

HULL.

The hull is constructed of mild steel, the outer plating being composed generally of 15-pound plating. There are one hundred transverse frames, spaced 3 feet apart, and formed of

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Z-bars (frames 18 to 36), 6 inches by 3½ inches by 3½ inches by 15½ pounds, except forward and aft (frames 1 to 17 and 87 to 100), which are 4 inches by 3½ inches by 9.1 pounds angle bars. The outside plating is sheathed with Georgia pine, 4 inches thick, extending to about 42 inches above the water-line amidships, shearing up forward to 69 inches, and aft to 51 inches.

Length L. W., feet	292
over all, feet and inches	308-9
Beam molded, feet and inches	43-3½
extreme, feet	44
Mean draught (with 467 tons coal and two-thirds stores), feet and inches	15-9
Corresponding displacement, tons	3,200
Area of midship section, square feet	615.2
load-water plane, square feet	9,370
Tons per inch of immersion	22.3

ARMAMENT.

Main battery	10 5-inch R. F. guns, 8 6-pdrs.
Secondary battery	2 1-pdrs., 4 Colt's automatic.
Mean height of axis of 5-inch guns, main deck, above load-water line, feet and inches	21-5
gun deck, above load water line, feet and inches	12-4

MACHINERY.

There are two triple-expansion, four-cylinder engines, of the vertical, inverted type, placed abreast of each other in separate watertight compartments, and designed to develop a total of 4,500 I.H.P. at about 172 revolutions. The engines are right-and-left, turn inboard, and have the following sequence of cylinders, beginning forward: Forward low pressure, high pressure, intermediate pressure and after low pressure.

Steam is supplied by six water-tube boilers of the Babcock & Wilcox *Alert* type.

ENGINES.

Number of engines	2
cylinders, each engine	4
Stroke (common), inches	30
Diameter H.P. cylinder, inches	18
I.P. cylinders, inches	29

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Diameter L.P. (2) cylinders, inches	35½
Volumetric cylinder clearances (mean of top and bottom) cubic ins.:	
H.P. (20.66 per cent.)	1,577
I.P. (15.87 per cent.)	3,143
L.P.'s (11.97 per cent.)	3,553
Valves, H.P. (one piston for each cylinder), diameter, inches	11
I.P. (two for each cylinder), diameter, inches	11
L.P. (one double-ported slide for each cylinder), inches, length, 25¾; width, 39½; port, length	37

DETAILS OF VALVE SETTINGS.

	H.P.		I.P.		L.P.	
Travel of valve, inches	4		4		4	
Side of valve on which steam is taken,	Inside.		Outside.		Outside.	
	Top.	*Bottom.*	*Top.*	*Bottom.*	*Top.*	*Bottom.*
Width of port, inches	1½	1½	1½	1½	1½×37	1½×37
Steam lap, inch	15/16	29/32	31/32	29/32	29/32	27/32
Exhaust lap, inch	−7/32	−1/32	−5/16	−1/16	+1/16	+3/32
Angular advance, degrees	35		35		40	
Steam lead, angular, degrees	7	8	6.25	7.75	13	14.50
linear, inch	7/32	5/32	6/32	8/32	11/32	7/16
Cut-off, in inches, maximum	23⅓	20¾	22 15/16	20⅝	22⅝	20
decimal of stroke	.771	.695	.764	.688	.745	.665
Exhaust release, in inches	27	26⅝	26⅝	26⅝	27¼	26¼
Compression, in inches	2 3/16	2	1⅞	1 15/16	4 3/16	3⅝
decimal of stroke	.072	.067	.060	.065	.138	.123
Steam opening, inches	1 1/16	1 3/32	1 1/32	1 3/32	1 3/32	1⅛
Exhaust opening	Full port.		Full port.		Full port.	

PROPELLERS.

Number of blades	3
Diameter, feet and inches	10–6
Pitch, feet and inches (as set on trial)	11 8 5/16
Helicoidal area, each screw, square feet	33.7
Projected area, each screw, square feet	28.7

BOILERS.

Number	6
Furnaces in each	Full width of boiler, with three doors.
Width of furnaces (internal), feet and inches	7– 7
Length of grate, feet and inches	6– 6
Heating surface (total square feet)	13,200
Grate surface (total square feet)	300
Working pressure (designed), pounds per square inch	275

MACHINERY WEIGHTS IN DETAIL.

Actual weights of the various parts of the finished machinery, as built and installed, are as follows:

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Group.			Tons.
I.	Main engine cylinders, receiver pipes, etc		25.486
II.	Shafting		19.660
III.	Main engine framing and bearings		25.698
IV.	Main engine reciprocating parts		7.403
V.	Main engine valve gear		6.845
VI.	Main condensers		11.583
VII.	Main air and circulating pumps and engines		5.584
VIII.	Propellers		5.692
IX.	Boilers		95.395
X.	Boiler fittings		38.050
XI.	Smoke pipes and uptakes		26.796
XII.	Steam and exhaust pipes and valves		15.390
XIII.	Suction and discharge pipes and valves		22.411
XIV.	Lagging and clothing		5.791
XV.	Flooring, gratings, etc		16.209
XVI.	Auxiliaries		20.748
XVII.	Fittings and gear		13.014
XVIII.	Water		36.718
XIX.	Stores, tools and spare parts		17.416
XIX*a*.	Government stores	19.709	
XX.	Miscellaneous machinery, heating, distilling and refrigerating plants		24.483
XXI.	Other miscellaneous machinery, C. & R. and Equipment	89.572	
XXII.	Connections under steam engineering to other miscellaneous machinery		3.267
	Total		443.639

COMPLEMENT.

Prior to the official trial, and as the *Chattanooga* neared completion, she was placed in commission at the navy yard, New York, on October 11, 1904. The trials were run, therefore, by her regular crew. Her complement of officers and men is—

Commissioned and warrant officers,	18
Deck force,	162
Artificer (deck) branch,	16
Engineer force,	83
Yeoman (special branch),	11
Messmen,	24
	314

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PROGRESSIVE RUNS TO STANDARDIZE SCREWS.

For the purpose of establishing the official speed of the vessel, as required by the contract, the standardized screw method was employed. The measured-mile course in Narragansett Bay, off Newport, R. I., was used to fix data necessary to construct the curve of speed and revolutions, shown on Plate 1 (on a reduced scale), from which later the speed of the vessel in free route and in the open sea for four hours was determined.

Instructions for standardizing the screws over the measured mile course called for runs alternately with and against the tide, as follows: Three single runs, at 16½ knots, or the highest attainable speed; two single runs, at each of the speeds of 16, 14, 12, 10 and 8 knots, or a total of thirteen single runs.

Table I contains, in tabulated form, the important data of the actual runs made, and Table II other machinery data calculated or observed. Two Weaver recording instruments (for description see page 86, Vol. III, of the JOURNAL), independently connected and located in the pilot house, were used to determine the revolutions and time on the measured mile at each run. An observer was stationed on the forward bridge and another on the after bridge, each of whom, by means of a push button, recorded on the tape of the Weaver instruments the instant of passing the ranges at each end of the mile course. These observers were also provided with stop watches, and the time on the course for each run was thus, in addition, ascertained. Column 2 of Table I shows the mean of the four observations for time (two Weaver instruments and two deck observers), and columns 3 and 4 the average revolutions for each shaft, as determined from the Weaver tapes. In all cases there was a close agreement.

Previous to the vessel starting for standardization trials, the draught was taken and found to be:

Forward, feet and inches, ...	15–3
Aft, feet and inches, ...	16–1
Mean draught, feet and inches, . . .	15–7
Corresponding displacement, tons, . .	3,156

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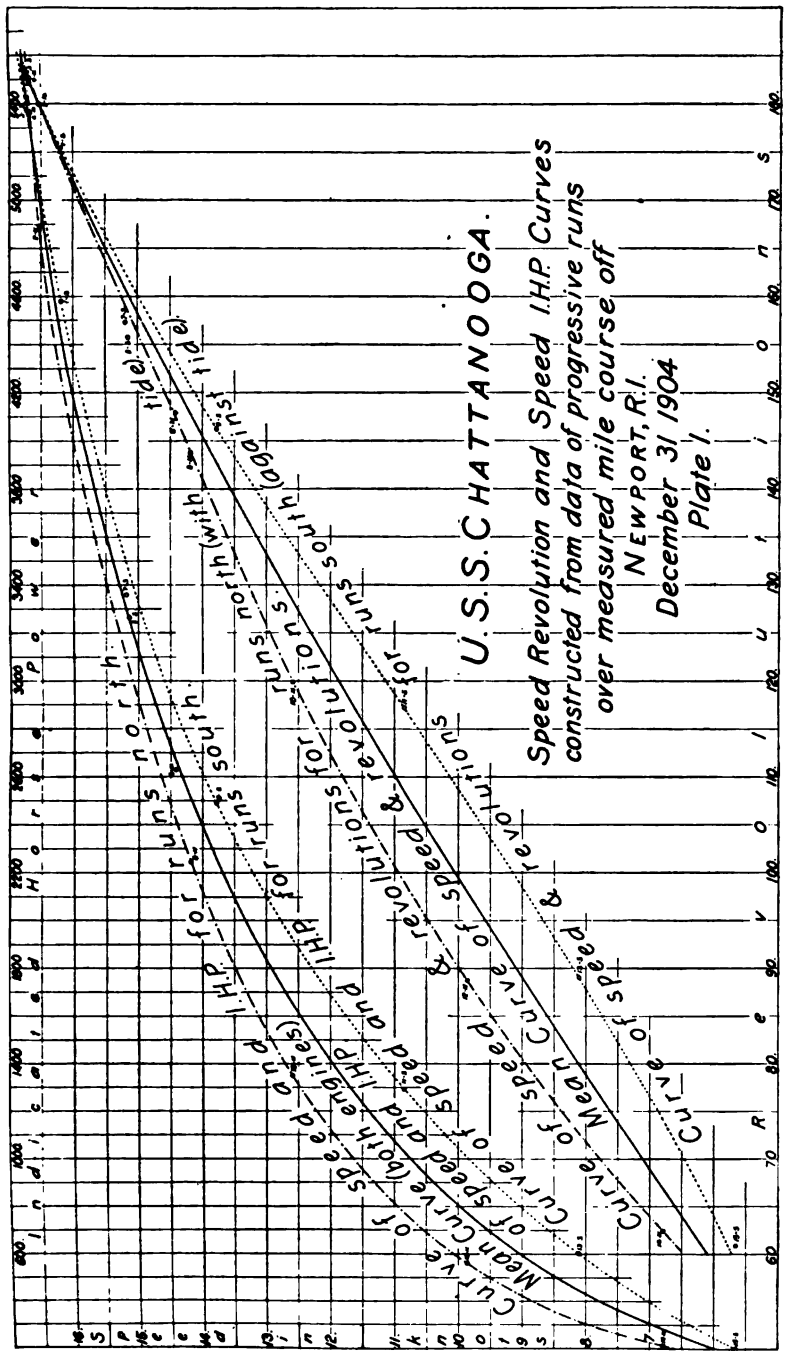

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TABLE I.

U. S. S. *CHATTANOOGA.*

Data from Progressive Runs over Measured-Mile Course off Newport, R. I., December 31, 1904.

Number and direction of run.	Time on course.		Revolutions of main engines. To make 1 knot. Starboard.	Revolutions of main engines. To make 1 knot. Port.	Revolutions of main engines. Per minute. Starboard.	Revolutions of main engines. Per minute. Port.	Revolutions of main engines. Per minute. Mean.	Speed in knots per hour.	Indicated horsepower. Starboard engine. H.P. cylinder.	Starboard engine. I.P. cylinder.	Starboard engine. Forward L.P. cylinder.	Starboard engine. After L.P. cylinder.	Starboard engine. Total.	Port engine. H.P. cylinder.	Port engine. I.P. cylinder.	Port engine. Forward L.P. cylinder.	Port engine. After L.P. cylinder.	Port engine. Total.	Total both main engines.	Auxiliaries in operation (estimated).	Total all machinery.
1	2		3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
	m.	*sec.*																			
1-s	3	42.35	655.1	645.35	176.78	174.14	175.46	16.19	700 5	802 3	418.9	401	2322.7	708.9	792	378	380.3	2259.2	4581.9	144	4725 9
2-n	3	37.17	650.1	649	179.11	178.81	178.96	16.53	698.4	876.6	401.3	386.3	2362 6	734.6	914.9	435.7	435.7	2520.9	4883.5	156	5039.5
3-s	3	35.18	655.3	655.95	182.72	182.9	182.81	16.73	730.9	896.6	460.2	440.1	2527.8	665.4	1116.7	530 5	530.5	2843.1	5370.9	164	5534.9
4-n	3	35.88	657.1	657.25	182.63	182.67	182.65	16.68	759.2	940.8	427.1	416.8	2543.9	796.9	1034.9	491 8	497.3	2820.9	5364.8	161	5525.8
5-s	3	57.63	617.45	616.2	155.9	155.59	155.75	15.15	538.6	576	202.2	242.8	1559 6	601.5	584 6	251.6	266.2	1703.9	3263.5	138	3401.5
6-n	4	09.6	615 6	615.15	147.98	147.88	147.93	14.42	456	460.9	165.1	192.6	1474.6	475	460.5	210.1	209	1354.6	2629.2	136	2765 2
7-s	3	57 05	619 65	623.4	156.84	157.79	157.32	15 19	515.5	562.2	227.5	288 2	1533 4	618.8	642.8	274.2	290.6	1826.4	3359.8	138	3497.8
8-n	4	13.93	607.65	608.95	143.58	143.9	143.74	14.18	382.4	412.6	156 1	156.1	1107.2	423.5	389.5	156.3	170	1139 3	2246.5	116	2382.5
9-s	4	21.25	634 85	638.55	145.8	146.65	146.23	13.78	408.4	443.3	160.7	170.9	1183.3	462 8	464	182.8	197.7	1307 3	2490.6	102	2592.6
10-n	4	45 95	571.05	571.55	119.82	119.93	119.88	12.59	226	214.8	113.7	109.3	663.8	241	203	121.5	121 5	687	1350.8	85	1435.8
11-s	5	31.95	649.25	648	117.35	117 13	117.24	10.85	214.8	188.6	104.7	101.4	609.5	233.2	208.4	119.2	122	682.8	1292.3	85	1377.3
12-n	6	05.03	539.25	540.45	88.64	88.83	88.74	9.86	101	82.2	45 1	43.7	272	110.5	87.8	50.4	41.4	290 1	562.1	74	636.1
13-s	7	24 03	661.75	656.95	89.42	88.77	89.1	8.11	112 9	101.1	49 5	45.4	308.9	111.1	87.8	47.9	45.4	292.2	601 1	74	675.1
14-n	8	50.13	532.75	531.6	60.3	60.17	60.24	6.79	43.9	29.5	14	17.9	105.3	43.4	33.5	15.1	16.2	108.2	213.5	55	268.5
15-s	10	36.33	638 9	635 6	60.24	59.93	60.09	5.66	40	28.3	14 6	15.2	98.1	47	29.6	15.1	13.9	105.6	203 7	55	258 7

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TABLE II.

U. S. S. *CHATTANOOGA*.

Data from Progressive Runs over Measured-Mile Course off Newport, R. I., December 31, 1904.

Number of run.	Time at beginning of run (approximate).	Which engine.	Steam pressures. In boilers above atmosphere.	Steam pressures. At engine above atmosphere.	Steam pressures. H.P. steam chest, absolute.	Steam pressures. In first receiver, absolute.	Steam pressures. In second receiver, absolute.	Mean effective pressures. H.P. cylinder.	Mean effective pressures. I.P. cylinder.	Mean effective pressures. Forward L.P. cylinder.	Mean effective pressures. After L.P. cylinder.	Mean effective pressures. Referred to low pressures.	Vacuum, inches.	Cut-off in cylinders from beginning. H.P.	Cut-off in cylinders from beginning. I.P.	Cut-off in cylinders from beginning. Forward L.P.	Cut-off in cylinders from beginning. After L.P.	Temperatures. Outside air.	Temperatures. Engine-room working platform.	Temperatures. Fireroom working level.	Temperatures. Injection.	Temperatures. Discharge.	Temperatures. Hotwell.	Temperatures. Feed.	Revolutions or double strokes. Circulating pump.	Revolutions or double strokes. Main feed pumps.	Revolutions or double strokes. Auxiliary feed pumps.	Revolutions or double strokes. Hotwell pump.	Revolutions or double strokes. Water-service pump.	Revolutions or double strokes. Bilge pump.	Revolutions or double strokes. Aux. cond'r. pump.	Revolutions or double strokes. Fireroom blowers.	Air pressure, in ins. of water
1-s	A. M. 9·08	St'b'd	257	235	230	93	30	106.1	45.9	15.93	15.25	44.16	25	.74	.73	.71	.71	36	88	70	33	72	60	60	240	26	...	...	60	90	...	420	1
		Port		245	230	79	30	109	46	14.59	14.68	43.61	24.5	.74	.73	.71	.71		79		34	74	78	86	172	14	40	16	26	46	49		
2-n	9·37	St'b'd	257	240	240	97	30	104.4	49.5	15.06	14.5	44.34	25.7	.74	.73	.71	.71	37	88	68	33	72	65	70	240	27	...	...	60	90	...	480	1
		Port		250	235	75	30	110	51.75	16.38	16.38	47.39	24	.74	.73	.71	.71		80		34	79	82	92	175	16	36	16	25	32	48		
3-s	9·58	St'b'd	263	250	250	103	29.5	107.1	49.63	16.93	16 19	46.5	25	.74	.73	.71	.71	38	88	66	33	72	64	64	190	30	...	90	60	90	...	508	1
		Port		262	260	89	36	97.4	61.75	19.5	19.5	52.25	23.5	.74	.73	.71	.71		81		36	70	76	86	260	21	30	16	24	74	48		
4-n	10·25	St'b'd	261	255	255	102	30	111.3	52.1	15.72	15.34	46.81	25.5	.74	.73	.71	.71	38	88	66	33	78	66	90	190	28	...	80	60	60	...	502	1
		Port		259	260	105	32.5	116.8	57.3	18.1	18.3	51.9	25	.74	.73	.71	.71		82		34	68	75	86	253	20	24	18	24	76	48		
5-s	10·45	St'b'd	269	250	195	75	24.5	92.5	37.37	8.72	10.47	33.63	27 5	.74	.73	.71	.71	39	88	72	33	78	65	90	180	30	...	90	50	60	...	435	1
		Port		265	185	59	21	103.5	38	10.87	11.5	36.8	26	.74	.73	.71	.71		80		34	60	68	78	246	18	30	18	24	52	44		
6-n	11·16	St'b'd	261	255	165	57	21	82.5	31.5	7.5	8.75	28.96	27	.74	.73	.71	.71	39	92	70	33	82	70	105	190	24	...	90	50	60	...	410	1
		Port		250	170	50	15.5	86	31.5	9.55	9.5	30.8	27.5	.74	.73	.71	.71		76		34	58	72	105	251	14	...	52	27	60	28		
7-s	11.37	St'b'd	242	235	165	60	20	88	36.25	9.75	9.78	32.87	28	.74	.73	.71	.71	39	92	68	33	72	65	110	190	24	...	100	50	60	...	400	1
		Port		225	185	78	22	105	41.2	11.68	12.38	38 9	27	.74	.73	.71	.71		78		34	58	60	110	239	16	...	64	24	52	24		
8-n	P. M. 12·10	St'b'd	248	240	155	55	17	71.3	29.06	7.31	7.31	25 92	28	.74	.73	.71	.71	39	90	72	33	72	80	100	180	26	...	60	40	50	..	390	1
		Port		242	175	50	16	78.8	27.38	7.3	7.94	26.61	27	.74	.73	.71	.71		76		34	57	74	100	242	16	...	40	26	54	28		
9-s	12 30	St'b'd	187	180	165	55	17	75	30.75	7.41	7.88	27.29	28	.74	.73	.71	.71	42	90	68	33	78	78	115	180	24	...	60	40	50	...	...	...
		Port		173	168	57	17	84.5	32	8.38	9 06	29.97	27	.74	.73	.71	.71		76		34	60	73	115	202	16	...	40	22	54	20		
10-n	1·00	St'b'd	245	230	145	40	18	50.5	18.13	6.38	6.13	18.63	28	.44	.65	.44	.44	45	88	68	35	72	70	115	170	20	...	96	40	60	...	...	...
		Port		237	140	37	15	53.8	17.12	6.81	6 81	19.25	27	.44	.6	.44	.44		76		34	48	78	115	246	14	...	58	26	68	28		
11-s	1·27	St'b'd	259	240	155	40	18	49	16.25	6	5.81	17.46	28	.44	.65	.44	.44	44	90	67	35	68	70	120	180	18	...	80	40	60	...	...	...
		Port		260	150	38	15	53.3	18	6.84	7	19.59	27	.44	.6	.44	.44		75		34	50	78	120	248	14	...	60	26	68	30		
12-n	2·16	St'b'd	258	240	90	25	9	30.5	9.38	3.42	3.31	10.32	28	.44	.65	.44	.44	43	86	68	35	65	95	140	160	16	...	70	45	40	...	...	...
		Port		264	73	18	8.5	33.3	10	3.81	3.13	10.97	26.5	.44	.6	.44	.44		77		34	43	100	140	252	14	...	64	28	68	32		
13-s	2·34	St'b'd	245	235	105	25	11	33.8	11.44	3.72	3.41	11.61	28	.44	.65	.44	.44	44	86	66	35	66	100	150	120	14	...	60	40	50	...	...	...
		Port		240	83	18	10	33.5	10	3.63	3.44	11.07	26.5	.44	.6	.44	.44		70		36	48	132	150	244	14	...	60	26	64	28		
14-n	2·55	St'b'd	234	230	55	15	3	19.5	4.94	1.56	2	5.87	28	44	.65	.44	.44	45	90	66	35	66	100	140	120	12	...	...	40	60	...	...	...
		Port		231	52	14.5	7	19.3	5.63	1 69	1.81	6 04	26	.44	6	.44	.44		75		36	42	131	140	232	12	...	...	26	62	28		
15-s	3·18	St'b'd	232	230	45	15	4	17 8	4.75	1 63	1.7	5.49	28	.44	.65	.44	.44	45	85	66	35	65	100	120	140	10	...	...	40	60	...	...	...
		Port		216	46	13.5	6.5	21	5	1.69	1.56	5.93	26.25	.44	.6	.44	.44		68		36	40	120	120	228	10	...	...	24	54	28		

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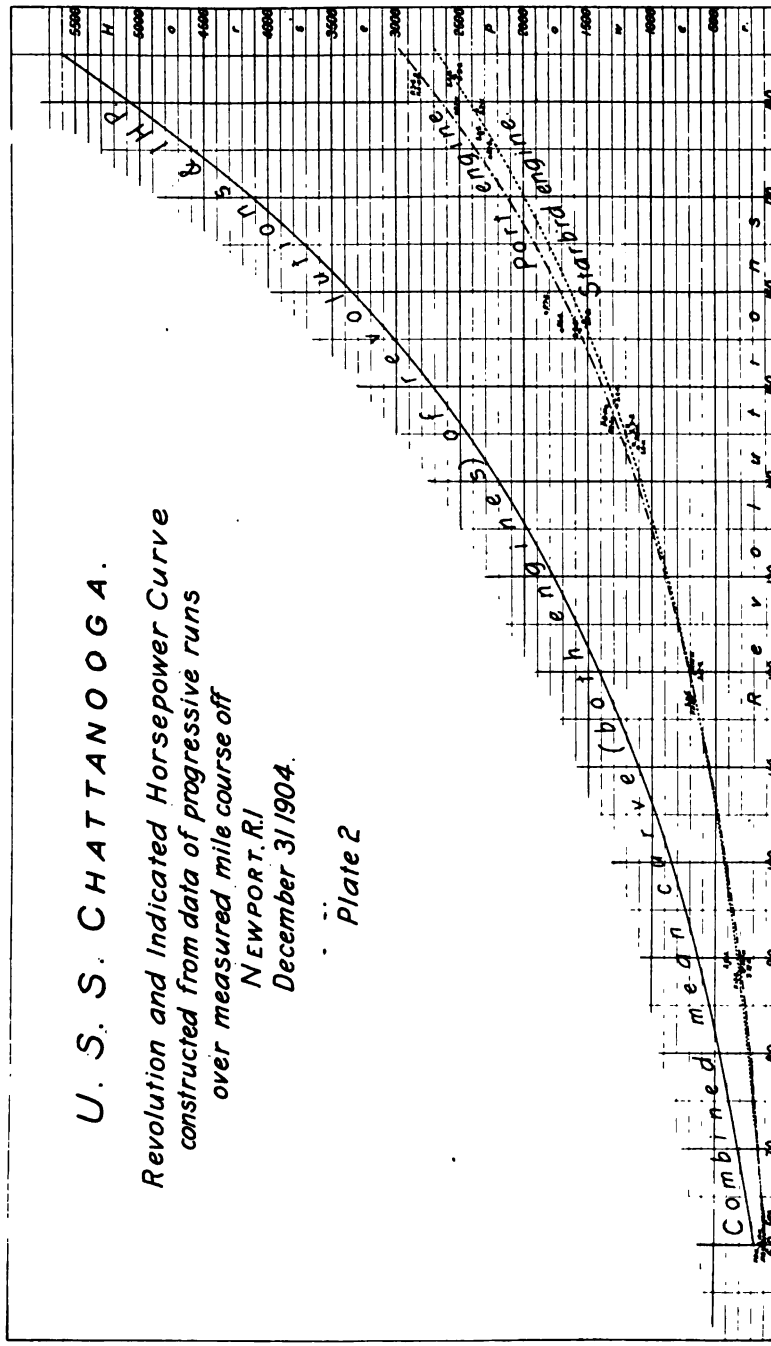

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After getting under way, water amounting to 44 tons was taken on board so as to insure a draught, when commencing the standardization runs, corresponding to a displacement of 3,200 tons.

The result of the trials with and against the tide (speed and mean revolutions) is plotted (dotted lines, Plate 1) as separate curves, and the curve of true speed constructed therefrom. From this latter curve, the deduction is made that an average of both engines of 179.7 revolutions per minute is requisite for a speed of 16.5 knots of the vessel.

On Plate 1 is also plotted curves of speed and indicated horsepower, and the mean curve obtained in the same manner as the mean curve for speed revolutions.

It is to be noted that points determined from runs Nos. 5 and 7, on the speed-revolution curve, fall so far without a fair curve running through other determined points as to indicate an abnormal condition when those runs were made. The cause is probably to be found in some diversity of the tide, due, possibly, to contour of the land, islands, etc., in the vicinity of the measured course.

Curves of revolutions and indicated horsepower, on the progressive runs are plotted and shown on a reduced scale in Plate 2. The curve for each engine (in dotted lines) is the mean for North and South runs, and the upper curve (shown in full) is the mean for both engines. From these curves, Tables III and IV have been constructed.

OFFICIAL FOUR-HOURS' CONTRACT TRIAL.

On January 1, 1905, at 9·15 A. M., the ship got under way from Newport, R. I., and proceeded to sea. The trial began at 11·50 A. M. and was finished at 3·50 P. M. The course steered was South 3 degrees West for two hours, when a long turn to the Westward was made and the course changed to North 3 degrees East. A moderate breeze was blowing from W. by N., with a moderate swell from the Southwest.

Before getting under way for the trial the mean draught was found to be 15 feet and 10 inches, corresponding to a displace-

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TABLE III.

U. S. S. *CHATTANOOGA.*

Data Deduced from Progressive Runs over Measured-Mile Course, off Newport, R. I., December 31, 1904.

Revolutions per minute.	Speed in knots per hour.	Indicated horsepower.			Slip of propeller in per cent. of its own speed.*	Constant †
		Starboard engine.	Port engine.	Total both engines.		
60	6.09	83	102	185	12.03	9.852
65	6.6	125	130	255	12	9.848
70	7.1	160	155	315	12.09	9.859
75	7.61	200	189	389	12.06	9.855
80	8.12	240	225	465	12.05	9.852
85	8.61	280	265	545	12.21	9.872
90	9.09	325	305	630	12.47	9.901
95	9.57	378	360	738	12.7	9.927
100	10.05	430	415	845	12.9	9.951
105	10.53	495	483	978	13.08	9.971
110	10.97	560	553	1,113	13.57	10.029
115	11.43	630	630	1,260	13.86	10.061
120	11.87	705	710	1,415	14.27	10.11
125	12.3	785	800	1,585	14.72	10.163
130	12.73	872	891	1,763	15.13	10.212
135	13.15	970	1,000	1,970	15.58	10.266
140	13.55	1,081	1,116	2,197	16.12	10.332
145	13.96	1,200	1,245	2,445	16.56	10.386
150	14.36	1,330	1,385	2,715	17.03	10.445
155	14.75	1,476	1,540	3,016	17.53	10.508
160	15.13	1,633	1,715	3,348	18.05	10.575
165	15.5	1,805	1,913	3,718	18.58	10.645
170	15.84	1,990	2,130	4,120	19.27	10.732
175	16.2	2,200	2,375	4,575	19.77	10.803
180	16.53	2,425	2,645	5,070	20.41	10.889
185	16.87	2,663	2,936	5,599	20.97	10.966

*Slip of propeller in per cent. of its own speed, based on mean (11 feet $8\frac{4}{16}$ inches) pitch

$$= \frac{(P \times R) - \left(S \times \frac{6{,}080.27}{60}\right)}{P \times R} \times 100.$$

† Requisite number of revolutions of the engines per minute to drive the ship one knot per hour.

ment of 3,225 tons. On anchoring at 6 P. M., after the trial, the mean draught was 15 feet $8\frac{1}{2}$ inches, and the corresponding displacement 3,189 tons.

During the trial the main machinery ran without appreciable vibration, and its performance was satisfactory. No water was used on any bearings, except the H.P. and I.P. crosshead journals of both engines, which ran slightly warm. All auxiliary machinery worked well with the exception of a derangement of the valve gear of one of the auxiliary feed pumps.

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TABLE IV.

U. S. S. *CHATTANOOGA.*

Data Deduced from Progressive Runs over Measured-Mile Course off Newport, R. I., December 31, 1904.

Speed in knots per hour.	Revolutions per minute.	Indicated horsepower.			Slip of propeller in per cent. of its own speed.	Constant, R + S.
		Starboard engine.	Port engine.	Total both engines.		
6	59	80	100	180	11.86	9.833
6.25	61.45	96	110	206	11.85	9.832
6.5	64	118	125	243	11.98	9.846
6.75	66.5	135	135	270	12.03	9.851
7	68.85	155	150	305	11.88	9.836
7.25	71.43	170	165	335	12.03	9.852
7.5	73.8	195	185	380	11.92	9.84
7.75	76.38	205	195	400	12.06	9.855
8	78.8	228	215	443	12.01	9.85
8.25	81.4	250	235	485	12.16	9.866
8.5	83.85	270	250	520	12.14	9.865
8.75	86.5	295	275	570	12.33	9.886
9	89.05	317	296	613	12.41	9.894
9.25	91.63	344	323	667	12.51	9.906
9.5	94.22	369	350	719	12.61	9.918
9.75	96.85	395	380	775	12.75	9.933
10	99.4	425	410	835	12.81	9.94
10.25	102.14	458	443	901	13.03	9.965
10.5	104.76	490	478	968	13.13	9.977
10.75	107.48	524	515	1,039	13.32	9.998
11	110 24	562	555	1,117	13.52	10.022
11.25	113	600	596	1,196	13.72	10.044
11.5	115.8	643	643	1,286	13.93	10.070
11 75	118.68	684	686	1,370	14.19	10.1
12	121.48	726	735	1,461	14.39	10.123
12.25	124.4	776	786	1,562	14.66	10.155
12.5	127.25	826	840	1,666	14.86	10.18
12.75	130.23	880	898	1,776	15.15	10.214
13	133.24	939	964	1,903	15.44	10.249
13.25	136.23	1,000	1,029	2,029	15.71	10.282
13.5	139.22	1,065	1,100	2,165	15.96	10.313
13 75	142.38	1,138	1,179	2,317	16.3	10.355
14	145.48	1,215	1,258	2,473	16.6	10.391
14.25	148.63	1,295	1,345	2,640	16.91	10.43
14.5	151.73	1,380	1,440	2,820	17.18	10 464
14.75	155	1,476	1,540	3,016	17.53	10.508
15	158.25	1 576	1,650	3,226	17.85	10.55
15 25	161.65	1,688	1,776	3,464	18.24	10.6
15.5	165	1,805	1,913	3,718	18.58	10.645
15.75	168.56	1,938	2,066	4,004	19.02	10.702
16	172.1	2,078	2,232	4,310	19.43	10.756
16.25	175.75	2,234	2,419	4,653	19.87	10.815
16.5	179.7	2,410	2,625	5,035	20.42	10.89
16.75	183.2	2,575	2,828	5,403	20.76	10.937

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The boilers steamed freely, and the automatic regulation of the firing, as well as the excellent organization of the fireroom force, contributed largely to the success of the trial. A timing apparatus for firing was improvised on board ship by Lieutenant A. T. Graham (the Chief Engineer), and proved so efficient in practical operation that a brief description may not be out of place. It consisted of a wooden annular ring, about an inch in width, placed on the face of a fireroom clock. In this, circumferentially, at equal intervals, fifty-four brass brads were driven, and every sixth one connected in circuit by wires set in the back of the ring. The minute hand of the clock had attached to it a small piece of wire which in revolving engaged, successively, the brass brads, and thus completed, when thrown in for operation, electrical contact through the dynamo circuit. A wooden box with transparent sides was hung between the firerooms, containing six 5-candlepower lights, arranged in compartments one above the other; when these lighted there was displayed consecutively the numbers 1, 2, 3, 4, 5, 6, which were continuously repeated at regular intervals, and which were visible to both firerooms. The furnaces—two boilers—in each fireroom were numbered from 1 to 6, and marked in an athwartship direction as shown:

BOILER B.		BOILER A.
2 - 5 - 3	*Forward Fireroom.*	4 - 6 - 1
1 - 6 - 4		3 - 5 - 2
BOILER D.		BOILER C.
BOILER F.	*After Fireroom.*	BOILER E.
2 - 5 - 3		4 - 6 - 1

From the above it will be observed that the firing interval was $1\frac{1}{3}$ minutes. The firemen were instructed to spread evenly over the grate six shovels-full of coal at each firing, and to fire only when the number of their furnace was displayed. In addition, a height of six inches above the grate was marked with aluminum paint on the furnace door front of each furnace, so that the height of the fire might be readily judged at any time. Instructions were given to fire less coal than the amount stated,

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if the fire exceeded six inches in height, and more if the reverse was the case. The only other instructions necessary were as to levelling the fires, which was always done (with a "devil's claw") just before firing each furnace. For example, when No. 1 fired and closed the furnace door, No. 2 immediately and quickly levelled and stood by to fire when his light came up, particular care being exercised, at all times, to keep furnace doors open no longer than absolutely necessary.

No slice bars were used during the trial.

The mean average revolutions for both engines, during the four-hours' run was 181.8, and the speed per hour, as deduced from the speed-revolution curve, 16.65 knots, or an excess of .15 knot over the contract requirements. Revolutions of each main engine were taken by two counters, separately connected, at 15-minute intervals; maximum and minimum revolutions for such periods are as shown below:

	Starboard.	*Port.*
Maximum revolutions, main engines	184.6	184.7
Minimum revolutions, main engines	179.0	179.5

PERFORMANCE.—FOUR HOURS' OFFICIAL TRIAL.

Steam Pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Steam pressure at boilers (per guage), pounds	269.4	
H.P. steam chest (gauge), pounds	238	238
1st receiver (absolute), pounds	100.6	105.8
2d receiver (absolute), pounds	31	36
Vacuum in condensers, inches of mercury	25.94	25.43

Temperatures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Injection, degrees	39.8	39.8
Discharge, degrees	80	77.6
Hotwell, degrees	80.4	77.1
Feed-water, degrees	130	117.6
Engine room, upper platform, degrees	101	101
working platform, degrees	83.3	75.1
Firerooms, above grating, degrees	94	
working level, degrees	88.6	

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Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

Main engines (mean for trial)	182	181.6
average for both	181.8	
Pumps, main, air	Attached.	
circulating	173.7	227.2
feed (forward and after)	27	18.1
hotwell	72.5	42.5
bilge	60	40.8
water service	50	20.3
auxiliary, feed	...	20
condenser	...	23.2
Blower engines, starboard, forward	493	
port, forward	497	
after	500.2	
Speed of ship, in knots per hour	16.65	
Slip of propeller, in per cent. of its own speed, based on mean pitch and average speed	20.71	20.57
Air pressure in firerooms, inch of water	1	

Mean effective pressures in cylinders, in pounds per square inch. (*Averages of cards taken at half-hourly periods.*)

Main engines, H.P. cylinder	116	115.56
I.P. cylinder	51.44	54.62
F.L.P. cylinder	15.58	16.22
A.L.P. cylinder	15.43	17.40
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	47.16	49.47

INDICATED HORSEPOWER.

Main engines, H.P. cylinder	788.5	783.8
I.P. cylinder	925.6	980.8
F.L.P. cylinder	421.8	438.2
A.L.P. cylinder	417.8	470.1
Total	2,553.7	2,672.9
both engines	**5,226.6**	

AUXILIARIES.

Circulating pumps, main	12.5	26
Feed pumps (main and auxiliary)	38.	
Hotwell pumps	5	3
Bilge pumps	1.2	.5
Water-service pumps	2.3	.7
Auxiliary condenser pumps	...	1.2
Blower engines (forced draft)	45.	
Steering engine (estimated)	3.	
Dynamo engine (one in operation)	31.	
Total	**169.4**	
Total all machinery in operation	**5,396.0**	

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COAL.

Kind and quality	Pocahontas, hand-picked, excellent.
Value in B.T. units (by calorimetric tests)	15,165
Pounds, per hour	10,632.1

DEDUCED DATA.

I.H.P. (total) per square foot of grate surface	17.99
heating surface	.4088
Pounds of coal per I.HP. per hour (main engines only)	2.03
all machinery in operation,	1.97
Pounds of coal per square foot of grate surface	35.44
heating surface	.8055
Cooling surface (main condenser), square feet per (total) I.H.P.	1.11

ENDURANCE TRIAL.

On completion of the four-hours' trial, it was the intention to make a 24-hour endurance run at about two-thirds horse-power, to determine reliable data concerning speed, coal consumption, and general efficiency of the machinery. For this purpose, the vessel got under way from Newport, R. I., about 1·30 P. M., January 3, 1905, and the trial began at 2·30 P. M. The weather was cloudy and rainy, and the wind fresh and increasing from the Northeast, with a moderate sea from the same direction. By 11 P. M. the wind had increased to a moderate gale, with a corresponding increase in sea, making it necessary to slow down, both on account of the amount of water taken over the bow and the thickness of the weather. The trial was, therefore, abandoned. Important data collected during eight hours of this trial is given below:

PERFORMANCE—ENDURANCE TRIAL.

Date and hour	January 3, 1905, 2.30 to 10.30 p. m.		
Number of hours		8	
	Starboard.		*Port.*
Mean revolutions of engines	157.9		157.5
both engines		157.7	
Steam pressure in boilers, pounds, per square inch,		255.	
at engines, pounds, per square inch,		244.	
H.P. steam chest, absolute	219.5		237.5
1st receiver, absolute	87.5		98.5
2d receiver, absolute,	31		35.5
Steam cut-off H.P. cylinder from beginning	.52		.58

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Steam cut-off I.P. cylinder from beginning	.59	.60
F.L.P. cylinder from beginning	.47	.52
A.L.P. cylinder from beginning	.47	.47
Vacuum, in inches of mercury	26.5	26.75
TEMPERATURES.		
Injection	44	
Discharge	78	77
Feed-water	143.5	135
MEAN EFFECTIVE PRESSURES.		
Main engines, H.P. cylinder	77	82.4
I.P. cylinder	34.2	36.7
F.L.P. cylinder	12.82	16.25
A.L.P. cylinder	12.3	14.11
M.E.P. referred to L.P.'s	33.6	37.73
INDICATED HORSEPOWER.		
Main engines, H.P. cylinder	454.1	484.8
I.P. cylinder	534	571.5
F.L.P. cylinder	301.1	380.8
A.L.P. cylinder	288.9	330.6
	1,578.1	1,767.7
Total, both engines	**3,345.8**	
I.H.P. of all auxiliaries in operation	**122**	
Total, all machinery in operation	**3,467.8**	
Grate surface in use	300	
Pounds of coal used per hour	5,947.4	
per square foot of grate,	20.12	
per I.H.P. per hour (main engines),	1.78	
all machinery in operation	1.72	
Mean draught at beginning of trial, feet and inches,	15–11½	
speed, by curve, in knots per hour	14.93	
Slip of screw, per cent	17.95	
Average air pressure, in inches of water	.5	

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NOTES.

WARSHIP BUILDING IN 1904.

Largely owing to reduced production from the private yards, the tonnage of warships launched during the year now closing is considerably less than in the previous year, and is much below the average of the past seven years. During that period the fighting ships launched from all British yards, for foreign navies as well as our own, made up a total of over 1,000,000 tons, or an average of 145,000 tons per annum; and the forty vessels floated during 1904 make up collectively a displacement tonnage of only 127,452 tons.. So far as the dockyards are concerned, the output is well up to the average, notwithstanding that extensive work has been carried out in connection with repairs and overhauls. The reduction in the work allotted to private firms is very marked, the tonnage launched being little more than one-half what it was for the British Navy in 1903, while in 1901 the total was more than double the tonnage of 1904. This question of the relative proportion of work allotted to the Royal establishments and to private firms has aroused attention during the past year, and the figures seem to justify the grievance expressed by such firms who have laid themselves out to satisfy Admiralty requirements, both in respect to efficient workmanship and time of delivery, and who are now short of naval orders. In 1903 three battleships were to have been ordered from contractors; but these at the last moment were given to the dockyards, the promise being made that there would be compensatory work for private firms in the succeeding year. But during the twelve months now closing only two battleships have been ordered from private firms, and no cruisers, the dockyards getting three cruisers. It is generally accepted that new work can be more economically carried out by contract than in the Government works, for the unavoidable reason that the conditions required

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for repair work are inimical to economical new construction. In the one case the plant requires to be distributed over a wide area, so that machine work for ships in docks, often widely separated, can be carried out close to the vessel so as to reduce handling; whereas with new construction the machine tools are preferably concentrated at the building berths. This

TABLE I.—FIGHTING VESSELS LAUNCHED.

.....	No.	Tons.	Indicated horse-power.	Value of ships completed.
In 1904, dockyards	4	57,100	80,500	£4,660,000
private yards (H.M.S.)	31	67,552	279,500	6,015,000
(foreign)	5	2,800	3,200	62,000
Total in 1904	40	127,452	363,200	10,737,000
In 1903, dockyards	4	28,290	42,800	2,384,000
private yards (H.M.S.)	31	120,456	277,300	9,848,000
(foreign)	7	4,237	17,190	368,000
Total in 1903	42	152,983	337,290	12,600,000
In 1902, dockyards	5	51,560	77,000	3,723,000
private yards (H.M.S.)	10	40,494	108,000	3,208,000
(foreign)	7	990	8,550	145,000
Total in 1902	22	93,044	193,550	7,076,000
In 1901, dockyards	8	64,910	114,300	4,901,200
private yards (H.M.S.)	24	144,190	275,000	11,002,600
(foreign)	7	2,442	47,500	340,000
Total in 1901	39	211,542	436,700	16,243,800
In 1900, dockyards	4	5,230	11,200	394,600
private yards (H.M.S.)	17	30,374	125,800	2,531,600
(foreign)	8	25,827	42,750	1,925,000
Total in 1900	29	61,431	179,750	4,851,200
In 1899, dockyards	6	66,900	78,000	4,901,100
private yards (H.M.S.)	12	53,222	111,000	3,791,000
(foreign)	16	47,170	124,000	3,767,000
Total in 1899	34	167,292	313,000	12,459,100
In 1898, dockyards	8	70,955	84,800	4,441,000
private yards (H.M.S.)	22	70,033	168,800	4,242,000
(foreign)	18	52,365	144,250	3,480,000
Total in 1898	48	193,353	397,850	12,163,000

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partly explains the lack of economy; there is, too, the difference in working hours, and in the amount of work done by the Government workers, as compared with that by private employés. The time is coming when it may be advisable to give all new warship work to contractors. The saving to be effected would be from 10 per cent. to 15 per cent., which in the aggregate would quite equal each year the cost of one of the largest type of destroyers.

But to return to the work of the year, it will be seen from Table I, that thirty-five ships for the King's Navy were launched from the various yards throughout the kingdom. These have a collective tonnage of 124,652 tons, while the indicated horsepower of the machinery aggregates 360,000. In the previous year there were thirty-five vessels, making up a total of 148,746 tons, the horsepower being 320,100. There is thus a decrease of 14,000 tons, but owing to the inclusion this year of a larger number of high-speed destroyers, the indicated horsepower is about 40,000 greater; 1902 was a lean year, the tonnage launched then being only 92,000. The best year in recent times was 1901, when the total was 209,000. Fewer large ships were floated this year, the list including only two battleships; whereas there were six in 1903; five cruisers, all of the armored type were floated, as compared with four in the previous year; two third-class cruisers, six scouts, one gunboat thirteen torpedo-boat destroyers, a torpedo boat, three improved and enlarged submarines, and two coast-guard cruisers, making up a total of thirty-five, including every type of the modern fleet.

As shown in Table II, the dockyard tonnage is 10,000 tons above the average of the preceding fourteen years; but at the present time the tendency is to reduce the staff at the Royal establishments, and we hope, for reasons we have already expressed, that the time is coming when these concerns will be confined to repair work. In the previous year a large amount of such repair work was given out to contract, ships as a rule being sent to the works where they were originally built; but, owing to the extreme difficulties of preparing a comprehensive esti-

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mate and of subsequent accounting, the financial arrangements have not, it is said, been satisfactory to the Treasury, so that from all points of view the dockyards would be more suitably employed in carrying out such repair work than in new construction. In the United States the experiment was made of building in competition a battleship in one of the Government yards and a second in private works, and the result, as might have been anticipated, was so much in favor of contract work that the former practice of giving no new work to Government establishments will be maintained.

TABLE II.—FIFTEEN YEARS' PRODUCTION OF BRITISH NAVY SHIPS.

Year.	Dockyard.		Private yard.		Total.	
	No.	*tons.*	*No.*	*tons*	*No.*	*tons.*
1890	8	22,520	13	42,475	21	64,995
1891	8	68,100	10	39,150	18	107,250
1892	9	50,450	13	90,750	22	141,200
1893	9	32,400	5	1,919	14	34,319
1894	8	26,700	19	4,825	27	31,525
1895	8	70,350	28	66,412	36	136,762
1896	9	70,970	26	36,515	35	107,485
1897	4	31,885	22	34,111	26	65,996
1898	8	70,955	22	70,033	30	140,988
1899	6	66,900	12	53,222	18	120,122
1900	4	5,230	17	30,374	21	35,604
1901	8	64,910	24	144,190	32	209,100
1902	5	51,560	10	40,494	15	92,054
1903	4	28,290	31	120,456	35	148,746
1904	4	57,100	31	67,552	35	124,652
Totals.....	102	718,320	283	842,478	385	1,560,798

Of a total average of about 104,000 tons per annum the private yards contribute 56,100 tons and the dockyards 47,900 tons. The dockyards, however, are at the moment very fully occupied, the four yards now engaged on new construction work having yet to float 90,050 tons, as compared with 114,000 tons a year ago and 68,390 tons two years ago. Portsmouth constructed the only two battleships included in this year's total, while, of the two armored cruisers, Chatham launched the *Devonshire* and Pembroke the *Duke of Edinburgh*. Devonport did not float any vessels, but has a battleship in a fairly advanced state, and a cruiser is being laid down. Chatham has similar ships in hand; and at Pembroke two armored

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cruisers are in hand, one of them almost ready for launching. A better method of indicating the extent of work done is, perhaps, to take the production of the yards over a period of years, and this is done in Table III.

TABLE III.—THE PRODUCTION FROM EACH NAVAL YARD.

Yard.	1904.		1903.		Total for 15 years.	Average for 15 years.
	No.	*tons.*	*No.*	*tons.*	*tons.*	*tons.*
Portsmouth	2	32,700	1	9,800	206,131	13,742
Chatham	1	10,850	...	...	191,212	12,747
Devonport	...	...	1	16,350	148,768	9,917
Pembroke	1	13,550	...	...	141,470	9,431
Sheerness	...	...	2	2,140	31,299	2,086

The work done by private yards is much less than in several previous years, excepting 1902 and 1900. This, of course, is taking into account the foreign work done, the only vessels launched this year, other than those for the British Navy, being a Canadian cruiser built by the Vickers Company, and three boats constructed by Messrs. Yarrow. But there is on hand, fortunately, a greater amount of foreign warship tonnage than has been the case for three or four years. The total tonnage of the warships built in private yards this year is only 70,352, whereas in the previous year it was 124,700 tons; in 1901, 146,600 tons; in 1900, 56,000 tons; in 1899, 100,000 tons; and in 1898, 122,000 tons. These figures speak for themselves, and indicate that if firms are to be encouraged to retain berths for warship work—which berths require to be of specially sound character, owing to the concentration of load in short, beamy battleships and cruisers—then the Admiralty should offer greater consideration to the question of the respective claims of dockyards and contractors.

As to the design of the battleships built we do not propose to enter into details, especially as we have in past issues referred to the widely-entertained opinion that many of our modern ships are weak in ordnance, because they have too many guns of small caliber, recent war operations having shown that the strongly-held opinion in favor of long-range

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guns is correct. The two battleships belong to the *King Edward VII* class, of which five were launched in the preceding two years. Of these, four have now completed their trials, and in all cases the legend speed of 18½ knots has been exceeded by practically half a mile per hour. To these and other trials we shall refer in a subsequent article. There are two more of the class yet to launch—the *Hibernia* and the *Africa*—building respectively at Devonport and Chatham. There will thus be eight vessels of the same class, whose efficiency would be enormously enhanced by the substitution of more 9.2-inch or of some 7.5-inch guns in place of the ten 6-inch weapons now fitted. They have a displacement tonnage of 16,350 tons and engines of 18,000 horsepower. The dockyard ships will cost over £1,400,000, or £60,000 more than the sister ships built by the contractors.

Of the five cruisers launched, the *Devonshire*, floated at Chatham, the *Roxburgh*, launched by the London and Glasgow Company, and the *Argyll*, by Scott's Shipbuilding and Engineering Company, of Greenock, belong to the improved County class, with 6-inch armor on the broadside, having a total displacement of 10,850 tons, and machinery of 21,000 horsepower, to give a speed of 23 knots. The other two cruisers—the *Black Prince* and the *Duke of Edinburgh*—are the first vessels designed by Mr. Philip Watts as Director of Naval Construction, and are considerable advances upon even the improved County class. But Mr. Watts, in his *Minotaur* class, has done very much better, as in them the 6-inch gun has been entirely discarded, the armament consisting of four 9.2-inch guns and ten 7.5-inch guns, whereas in the *Black Prince* there were six 9.2-inch and ten 6-inch guns. We recently entered at length into the question of the gun power of our cruisers, and we need not enlarge upon the serious defects of many of the vessels not yet completed.

Little requires to be said about the third-class cruisers, of which two were built similar to two launched in the preceding year. These four vessels—the *Diamond* and *Topaze*, built by Messrs. Cammell, Laird & Co.; the *Sapphire*, built by Palmers'

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Shipbuilding and Iron Company; and the *Amethyst*, built by Sir W. G. Armstrong, Whitworth & Co.—have all completed their trials; and as the *Amethyst* was fitted with Parsons steam turbine, opportunity was afforded of making a comparison of this newer motor against the ordinary reciprocating machinery fitted in the others. There was the minimum number of variants, so that a fairly safe deduction could be made as regards economy, and the results—which we published in considerable detail some weeks ago—showed clearly that for practically all powers the turbine machinery is more economical, the difference being greatest at the highest rates of speed, when the steam consumption of the turbines was about 30 per cent. less than with the reciprocating engines.

Six of the new 25-knot scouts have been launched. Of these, two—the *Pathfinder* and *Patrol*—were built by Messrs. Cammell, Laird & Co., Birkenhead; two—the *Forward* and *Foresight*—by the Fairfield Company; one—the *Sentinel*—by the Vickers Company, and the sixth—the *Adventure*—by Sir W. G. Armstrong, Whitworth & Co. The two last-named firms have each a sister ship to launch, but are very properly awaiting the results of trials. The *Sentinel* is the only one completed, and will proceed on her final trials within a few weeks. Already some experimental tests have been made which have given very promising results. The design of the machinery of these vessels is of very considerable interest, because of the limitation in weight, notwithstanding the 17,000 horsepower to be developed. In some of them very light torpedo-destroyer machinery has been adopted, and Messrs. Hawthorn, Leslie & Co., who are engining the Elswick boats, are fitting on each shaft two independent sets of triple-expansion engines, making six cylinders in all. In the Vickers ship the engines are substantial, and resemble more the cruiser type, so that a comparison of the results of the trials of all ships, and especially of the endurance of the machinery, will certainly be of considerable interest.

Thirteen destroyers were launched, all of the new 25½-knot class. Three of these—the *Ure*, *Wear* and *Rother*—were built

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by the Palmer's Company; three—the *Boyne*, *Doon* and *Kale*—by Messrs. Hawthorn, Leslie & Co.; two—the *Ribble* and *Welland*—by Messrs. Yarrow; two—the *Liffey* and *Moy*—by Messrs. Cammell, Laird & Co.; two—the *Jed* and *Chelmer*—by Messrs. Thornycroft; and one—the *Ness*—by Messrs. J. S. White & Co., of Cowes. Their displacement was from 580 to 600 tons. These vessels, which have high forecastles, were undoubtedly ordered under a misconception. Following upon the accident to the *Cobra*—a vessel of 30 knots—a court-martial raised doubts, on imperfect knowledge, as to the strength of the 30-knot destroyers, and for this reason scantlings and weights were increased, with the result that speed had to be reduced, and now even 25½ knots is only attained with considerable difficulty.

A committee of experts, appointed subsequently to the court-martial, made a most searching investigation, and a series of interesting experiments at sea with 30-knot boats, and their report, which unfortunately has not yet been issued, is understood to prove that the doubts raised by the decision of the courtmartial were baseless. Now the Admiralty propose to construct destroyers to attain a speed of 32 to 33 knots. The builders invited to tender have been left a free hand as to the design and the system of propulsion; but it would almost appear that the turbine is practically essential to the attainment of this very high rate of speed. In view, however, of the fact that at low power considerable difficulty is experienced in attaining a high economy in turbine-driven destroyers, it is probable that in high-speed turbine craft reciprocating engines will be fitted forward of the turbines for cruising purposes, and perhaps also for going astern; but it is yet too early to indicate the exact nature of the proposals put forward for the consideration of the Admiralty. A torpedo boat was also built by Messrs. J. S. White & Co., of Cowes, and the shallow-draft gunboat *Widgeon* by Messrs. Yarrow. The coastguard ships by Messrs. Workman, Clark & Co., of Belfast, and by Messrs. Bow, McLachlan & Co., of Paisley.

Three more submarines have been added to our fleet from

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the Vicker's Company's works, but patriotism justifies our referring to these only in very general terms. It is very satisfactory to know that the resources of this important constructional firm—perhaps the most progressive in the country—have been placed unreservedly and exclusively at the disposal of the Admiralty in this matter; and the best results are the more likely to be achieved by the ready co-operation between the inventive engineers of the company and the officers of the Navy most qualified for this by experience and knowledge of the problems of submarine navigation. The consequence of practical work with these new engines of naval warfare is that the British type of boat has many qualities which, it is understood, are lacking in the boats of foreign navies. We have the assurance of the Admiralty that most satisfactory results are being achieved, and since secrecy in the design and equipment of these vessels is of the most vital importance to the maintenance of our naval power, all will regard this assurance, even in the absence of definite evidences, as meeting the special circumstances of the case.—"Engineering," London.

WARSHIP STEAM TRIALS IN 1904.

There were rather fewer warships passed through their steam trials last year than in the immediately preceding years; but the list, as we shall presently show, is of considerable interest, because of the comparisons which can be made. In all there were twenty-six vessels, as compared with twenty-nine in the previous year, eighteen in 1902, thirty-two in 1901, and twenty-five in 1900. The list includes five battleships, one armored cruiser, four third-class cruisers, a torpedo gunboat, which had been re-engined, a sloop, a shallow-draught gunboat, nine destroyers, and four torpedo boats; whereas in the previous year there were four battle ships and eight armored cruisers, with about the same number of small craft, and ten destroyers and five torpedo boats. The results of the trials of the larger ships are set out in Table I, and we would direct special attention to the fact that we are able this year to include the water

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consumption of the great majority of these vessels. The measurement of water on trials involves the expenditure of national money; but the result is an important element in arriving at the relative economy of different types of steam generators and of the machinery, and the Water-Tube Boiler Committee are to be commended for bringing about this change, so long desired by the engineering officers of the Admiralty and by engineers generally. In Table II the results of the trials of torpedo craft are given; in their case the limitations of space and weight, in association with the rapid rate of steam generation, precludes similar tests being made.

Of the five battleships on the list, one, the *Swiftsure*, was, with the *Triumph*, originally built for the Chilian Government, and the specification called for steam trials under Chilian conditions, so that, although the tests were made by Admiralty officials, the nature and duration of the trials differed from those usually provided for in British naval contracts. These were, however, of a sufficiently severe nature and duration to adequately test the steaming qualities of the ships. The *Swiftsure's* performance we have already analyzed and compared with those achieved by the *Triumph*, completed for commission by the Vickers Company, while pointing out their superiority in many of the combative elements, associated as they have proved to be with higher speed than that usually attained by British warships. It is gratifying to know that in service both of these ships are giving very satisfactory results, and that the Admiralty have, in the redistribution of the ships of the fleet, placed them, as they ought all along to have been placed, along with 19-knot vessels of the *Russell* class, to which they are superior in speed as well as in gunpower.

The other four battleships belong to the *King Edward VII* class, of which four more are being built. So far as the offensive and defensive qualities of these vessels are concerned we have already dealt fully with them, and are now only concerned with the results of the trials. The *Dominion* and one or two of the other vessels were tried under new conditions, closely approximating to those appertaining to war. These

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conditions we have already reviewed at considerable length. The fact that they are feasible was demonstrated, perhaps more effectually in the case of the *Dominion* than in the other ships.

One point of interest in the comparison is that while the *Commonwealth* and *Dominion* are fitted entirely with Babcock and Wilcox boilers, the other two ships of the class included in Table I have a proportion of cylindrical boilers for use when steaming at low power, the water-tube boilers being intended for service at the higher rates of speed. This change was in conformity with the suggestion of the Navy boiler committee; but experience has shown that it is not quite satisfactory, and later ships are now fitted exclusively with water-tube steam generators. The results of the first 30-hours' trial, at one-fifth of the full power, are, from this point of view, suggestive. It will be seen that the two ships using water-tube boilers for this trial proved more economical from the point of view of coal consumption than the other two. Thus the water-tube boilers of the *Dominion* gave an evaporation of 11.2 pounds of water per pound of coal; while in the case of the *Hindustan* the result was 10.3 pounds, and in the case of the *King Edward VII* 7.6 pounds.

On the second 30-hours' trial, at three-quarters of the designed power—which, in the Navy, is regarded as the power which should be maintained so long as the coal in the bunkers may last—the results are more uniform, but the *King Edward VII*, again compares badly, while the Vickers ship proves exceptionally satisfactory, especially in respect of speed. The two ships fitted exclusively with water-tube boilers have again a lower consumption than the others, the *Commonwealth* coming out at 1.67 pounds, the *Dominion* at 1.68 pounds, whereas the rate for the *Hindustan* is 1.76 pounds, and for the *King Edward VII*, 1.96 pounds. The evaporative efficiency in the case of the *Dominion* is highest, being 11 pounds of water per pound of coal; but as the steam consumption of the machinery was higher than in the other ships, there may be some other cause partly accounting for the figures. In the

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TABLE I.—RESULTS OF OFFICIAL STEAM TRIALS OF WARSHIPS MADE DURING 1904.

Name of vessel.	Type of ship.	Builders of ship.	Makers of machinery.	Length in feet.	Displacement.	Type of boilers.	Heating surface.	Grate area.	First 30-hours' trial.				Second 30-hours' trial.				Full power.			
									Indicated horsepower.	Speed, log.	Coal per I.H.P. per hour.	Water, all purposes, per I.H.P. per hour.	Indicated horsepower.	Speed, log.	Coal per I.H.P. per hour.	Water, all purposes, per I.H.P. per hour.	Indicated horsepower.	Speed. l = log. m = measured mile.	Coal per I.H.P. per hour.	Water, all purposes, per I.H.P. per hour.
					tons.		*sq ft.*	*sq ft.*		*knots*	*lbs.*	*lbs.*		*knots.*	*lbs.*	*lbs.*		*knots*	*lbs.*	*lbs.*
Swiftsure..	Battleship	Armstrong	Humphrys, Tennant & Co.	436	11,800	Yarrow	37,524	664	8,700	17.217	2.39	...	...	...	...	...	14,031	19.93 l.	...	...
Commonwealth...	Ditto	Fairfield Company	Fairfield Co.	425	16,350	Babcock & Wilcox	47,360	1,402	3,663	11	1.73	...	12,797	16 88	1.67	16 77	18,562	19.37 l.	1.83	18
Dominion	Ditto	Vickers Sons & Maxim	Vickers Sons & Maxim	425	16,350	Ditto	47,360	1,402.5	3,889	12.8	1.93	21.6	12,843	18.3	1 68	18.4	18,438	19.35 l	1.77	18.3
King Edward VII.	Ditto	Devonport	Harland & Wolff	425	16,350	Babcock and cylindrical	43,700	1,309	3,760	11.6	2.63	20	12,844	17.35	1.96	16.3	18,138	19.04 l.	2.17	17.7
Hindustan	Ditto	J. Brown & Co.	J. Brown & Co.	425	16,350	Ditto	47,564	1,364	3,718	11.8	1.94	20	12,926	17.65	1.76	17.5	18,521	19 8 l.	1.8	18.3
Cornwall.	First-class cruiser	Pembroke	Hawthorn, Leslie & Co.	440	9,800	Babcock & Wilcox	54,826	1,616.5	4,798	15.3	1.73	17.8	16,498	21.83	1.69	17.45	22,709	23.8 l.	1.94	17.9
Amethyst...	Third-class cruiser	Armstrong	Parsons (turbine)	360	3,000	Modified Yarrow	25,968	494	4,890†	18.2	1.71	15.6	7,300†‡	20.63	1.5	13.8	14,200†	23.63 m	1.72	13.4
Sapphire...	Ditto	Palmer's	Palmer's	360	3,000	Reed	26,010	496	5,012	18.47	2.34	19.8	7,281‡	20.68	2.22	19.8	10,200	22.45 m.	2.52	22.2
Diamond...	Ditto	Laird's	Laird's	360	3,000	Laird-Normand type	26,000	490	5,074	18	2.27	19	7,143‡	20	2.32	19.31	10,066	22.17 m.	2.98	21.6
Topaze.....	Ditto	Ditto	Ditto	360	3,000	Laird-Normand type	26,000	490	4,992	18.1	2.3	19.2	7,092‡	20.2	2.2	19.49	9,860	22.1 m.	2 65	20.2
Halcyon.....	Torpedo gunboat	Devonport	Re-engined and re-boilered; Cammell, Laird	250	1,070	Laird-Normand	13,530	240	1,280	12.6	2.6	...	1,288*	12.25	1.92	...	6,194	20.5 l.	...	...
Cadmus.....	Sloop	Sheerness	J. S. White	185	1,070	Niclausse	4,000	135 8	331	6	2.12	...	1,068	11	1.82	...	1,433	13.4 m	1.97	...
Widgeon....	Shallow-d'ft. gunboat	Yarrow	Yarrow	160	180	Yarrow	2,654	95 2	...	...	...	...	...	...	...	...	671	13 m.	...	...

* Trial, 12 hours' duration. † Estimated indicated horsepower. ‡ Trial, 8 hours' duration.

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case of the *Commonwealth* and *Hindustan* 10 pounds of water were converted into steam per pound of coal, whereas in the case of the *King Edward VII* the result is again less—8.3 pounds. As regards the full-power trial, the interesting fact is that in all cases the designed power was exceeded, being in three cases practically 500 over the legend; the propulsive efficiency also seems to have been high. The designed speed was 18½ knots, but with the relatively slight increase in power 19.3 knots was realized, a result which must be regarded as very favorable. The *King Edward VII* was consistent in her high coal consumption, the rate at full speed being 2.17 pounds; the others range between 1.77 pounds and 1.83 pounds per horsepower hour. The evaporative efficiency in the case of the Vickers ship is again higher, being 10.3 pounds, whereas in the *Commonwealth* and *Hindustan* it is 10.3 pounds and 10.2 pounds, and in the *King Edward VII* it is once more very low—8.15 pounds. It would, of course, be a mistake to place too much reliance on these comparisons; although the results are taken with very great care, there must necessarily be a margin of inaccuracy; but against this there is the fact that the ships occupy the same relative positions to each other on each of the trials.

Of the cruisers the *Cornwall* is the only armored cruiser included in the list. She is the last of the ten vessels of the County class to be completed. The first vessels of the class, it will be remembered, did not at first obtain their legend speed of 23 knots; but when new propellers were fitted, having a larger ratio of blade area per unit of diameter, a great improvement was manifested, and this has been maintained by the *Cornwall*, which, on her full-power trial, attained a speed of 23.8 knots for 22,709 indicated horsepower, or 700 in excess of the designed power. The evaporation on the low-power and on the 75 per cent. trial, was 10.3 pounds per pound of coal; but on the full-power trial it dropped to 9.25 pounds. The steam consumption of the machinery—which, by the way, is of the short-stroke type—was uniform at about 17¾ pounds.

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The four third-class cruisers in the list are remarkable for the inclusion amongst them of the first steam-driven turbine cruiser afloat. This vessel—the *Amethyst*—in all other respects closely resembles the *Sapphire*, *Diamond* and *Topaze*, so that the results are especially interesting. We have, however, so recently reviewed the performances of these four vessels that it is not necessary here to enter at length into the question. The figures given may vary slightly from those already published, as those given in Table I have been subjected to a more careful analysis than was possible on the ships themselves, but the general result is the same. At one-half power the turbines required only 15.6 pounds of steam per horsepower, as compared with an average of 19.3 pounds by the other three ships; at three-quarters power the steam consumption dropped to 13.8 pounds, as compared with 19.5 pounds; while at full power the steam consumption was only 13.4 pounds, as compared with a mean of 21.3 pounds. The coal consumption per unit of power was less for the turbine boat, probably because the steam evaporation per square foot of surface was not so high. For the same weight it was possible to develop 4,000 horsepower more than with the reciprocating engines, so that the speed was 1¼ knots greater—the *Amethyst* recording on her measured-mile performance 23.63 knots, while the other three vessels ranged between 22.1 and 22.45 knots.

Little need be said of the other three ships in the list. One is a torpedo gunboat—the *Halcyon*—built under the Naval Defense Act scheme in 1894. She was then fitted with locomotive boilers, which gave a speed of only 19 knots; but now, with water-tube boilers and new machinery, she has succeeded in increasing her power from 3,500 to 6,194, attaining a speed according to log of 20.5 knots. The *Cadmus* is a sloop, and the results generally compare with those attained on similar vessels; while the *Widgeon* is a shallow-draught gunboat by Messrs. Yarrow, for service on the China rivers. This vessel we have already described, and dealt also with her trials.

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TABLE II.—RESULTS OF OFFICIAL TRIALS OF TORPEDO-BOAT DESTROYERS AND TORPEDO BOATS MADE DURING 1904.

Name of vessel.	Makers of hull and engines.	Length.	Displacement.	Type of boiler.	Heating surface.	Grate area.	Four-hours' speed trial.		Four-hours' coal consumption.		
							Indicated horsepower.	Speed.	Indicated horsepower.	Speed.	Coal per I.H.P. per hour.
		ft.	*tons.*		*sq. ft.*	*sq. ft.*		*knots.*		*knots.*	*lbs.*
Cherwell	Palmers	225	550	Reed	15,514	319	7,250	25.603	7,207	25.38	2.34
Dee	Ditto	225	550	Ditto	15,514	319	7,306	25.504	7,206	25.413	2.28
Derwent	Hawthorn	220	550	Modified Yarrow	14,852	277	7,241	25.68	7,580	25.295	2.24
Waveney	Ditto	220	550	Ditto	14,852	277	7,307	25.626	7,574	25.356	2.19
Ribble	Yarrow	225	590	Yarrow	16,000	270	7,417	25.817	7,668	25.33	1.572
Welland	Ditto	225	590	Ditto	16,000	270	7,777	26.24	7,720	25.35	1.654
Kennett	Thornycroft	220	550	Thornycroft-Schulz	16,160	268	7,445	25.665	7,543	25.99	2.395
Jed	Ditto	220	550	Ditto	16,160	268	7,990	25.787	7,425	25.39	2.464
Eden	Hawthorn—hull Parsons—turbine	220	550	Modified Yarrow	14,852	276	...	26.229	...	25.287	...
114	White	165	200	White-Forster	6,218	109.9	3,034	25.154	2,969	24.803	2.15
115	Ditto	165	200	Ditto	6,218	109.9	3,047	25.187	3,011	24.835	1.97
116	Ditto	165	200	Ditto	6,218	109.9	3,001	25.22	2,999	24.878	1.95
117	Ditto	165	200	Ditto	6,218	109.9	3,016	25.115	2,975	24.779	2

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The nine torpedo-boat destroyers tried during the year belong to the 25½-knot class, regarding the design of which craft doubts have been expressed, because, although much stronger and having a wider radius of action, the reduction in speed is viewed as serious by many naval strategists. Ten of these vessels were tried in the previous year, so that nineteen have now been added to the fleet, and several still remain to be completed. The results embodied in the table are fairly uniform, although the range in power is considerable. The practice in these vessels is to make first an endurance trial, the coal consumption on which determines the load to be carried at full power. This in part may account in some cases for the higher power required on the full-speed run. The results of the trials of the vessels built by Messrs. Palmers are notably successful in respect of propulsive efficiency. During the past two years 25½ knots have been realized by five of their vessels with powers ranging between 6,957 and 7,306 indicated horsepower. Messrs. Hawthorne, Leslie's vessels on Table II show similarly good results. The Yarrow boiler in the Yarrow boats has given very economical results; and although in the case of the *Welland* the power is 500 indicated horsepower higher than in some of the other ships, the speed is better by from .6 to .75 of a mile. One of the destroyers included in the list—the *Eden*—is fitted with turbine machinery; but there is no no means here of accurately determining the power required.

The miles run at full speed per ton of coal number about the same as those for the same design of vessel with the same type of boiler, but fitted with reciprocating machinery, *i. e.*, 3.4 miles per ton. At low power the results with the turbine are not so satisfactory. These comparative results, however, are not conclusive, and further trials are being made. Meanwhile the Admiralty are arranging for the building of 32 and and 33-knot destroyers, and it seems probable that, at least at full speed, better results will be achieved with the turbine as a propelling agent than with reciprocating engines; but a

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method has yet to be devised whereby a good economy can be got at low power.

The four torpedo boats on the list resemble the type added to the Navy from time to time to maintain the number of this craft at their present standard. These vessels are of 25-knots speed, and in all cases this rate has been exceeded with the legend power. The Admiralty are at the present time inviting tenders for the construction of five more vessels of the same class.—"Engineering," London.

SHIP BUILDING.

From the returns just issued by Lloyd's Register for the quarter ending December 31, 1904, we see that there is (excluding warships) an increase in the tonnage of vessels now under construction in the United Kingdom. This increase amounts to 151,382 tons over the returns for the corresponding quarter of 1903, the figures being 1,049,860 tons for 1904 against 898,478 tons for 1903. The tonnage belonging to the quarter just ended are accounted for by 403 vessels, against 386 in the corresponding quarter of 1903. At the end of September last there were building 393 vessels, totaling 1,046,308 tons, so that the figures for the quarter just ended and those for the previous one are very nearly alike. Of the total number of vessels now in hand, 371 are for steam and 32 for sail. The present figures, however, when compared with the record total reached in September, 1901, show a reduction of about 363,000 tons. Of the vessels now under construction, the surveyors of Lloyd's Register have 315 under their supervision, the tonnage of which is 748,177; but, in addition to these, 43 vessels, of 110,763 tons, are being built abroad, with a view to classification in the society. The total now building under the supervision of Lloyd's Register is, therefore, 358 vessels, of 858,940 tons. During the quarter ending December 31, 1904, there were 177 steamships commenced, having a gross tonnage of 329,457, together with

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eight sailing vessels, with a gross tonnage of 1,350. Of vessels previously commenced, but on which no further progress has been made, there are fifteen steamships, with a tonnage of 85,488, and six sailing ships, with a gross tonnage of 914.

TABLE XV.—TONNAGE OF SHIPS LAUNCHED (INCLUDING WARSHIPS) IN EACH MARITIME COUNTRY.

.....	1901.	1902.	1903.	1904.
Austria-Hungary	27,413	23,361	28,848	28,125
Belgium	5,631	14,946	12,804	6,686
China	3,694	3,400	2,364	709
Denmark	26,056	27,148	32,059	15,859
France	232,404	236,635	123,528	124,845
Germany	284,063	257,885	245,084	247,167
Holland	35,856	73,973	59,594	61,404
Italy	88,329	62,288	57,389	58,678
Japan	37,208	36,481	41,004	33,577
Norway	36,875	37,878	41,599	50,469
Russia	39,890	52,579	45,404	8,143
Sweden	19,191	9,030	15,505	10,375
Other European countries	2,496	5,507	3,669	3,436
United States	481,138	396,433	450,090	409,403
Other countries	...	...	592	763
Colonies	28,134	28,819	34,690	30,965
Total foreign and colonial,	1,348,378	1,266,363	1,194,223	1,090,604
United Kingdom	1,736,708	1,521,698	1,342,508	1,332,337
Total for world	3,085,086	2,788,061	2,536,731	2,422,941

TABLE XVI.—THE NUMBER AND TONNAGE OF WARSHIPS LAUNCHED, 1892-1904.

.....	British.		Foreign.	
	No	*tons.*	*No.*	*tons.*
1892	30	151,157	53	157,744
1893	18	45,898	33	133,635
1894	31	32,971	27	87,050
1895	59	148,111	23	100,614
1896	55	163,958	37	167,833
1897	48	95,465	42	133,435
1898	41	191,555	50	175,318
1899	35	168,590	56	176,170
1900	29	88,364	70	192,100
1901	41	211,969	82	255,578
1902	23	94,140	80	191,166
1903	41	151,890	78	239,210
1904	37	127,175	65	307,831

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TABLE XVII.—TABLE SHOWING THE NUMBER AND TONNAGE OF VESSELS OF 100 TONS GROSS AND UPWARDS (EXCLUDING WARSHIPS) LAUNCHED IN THE PRINCIPAL COUNTRIES.

Year.	United Kingdom.	Austria-Hungary.	British Colonies.	Denmark.	France.	Germany.	Holland.	Italy.	Japan.	Norway.	United States.	Other countries.	Totals.	
	tons.	*tons.*	*tons.*	*tons.*	*tons.*	*tons.*	*tons.*	*tons.*	*tons.*	*tons.*	*tons.*	*tons.*	*No.*	*tons.*
1892	1,109,950	2,302	19,792	12,707	17,228	64,888	14,368	13,888	...	24,572	62,588	15,762	1,051	1,358,045
1893	836,383	7,435	17,089	10,719	20,337	60,167	1,339	10,626	1,132	16,552	27,174	17,788	846	1,026,741
1894	1,046,508	1,703	6,803	7,300	19,636	119,702	15,360	5,396	3,173	17,169	66,894	13,894	932	1,323,538
1895	950,967	7,371	10,381	10,982	28,851	87,786	8,292	5,603	2,296	12,873	84,877	7,881	880	1,218,160
1896	1,159,751	6,246	11,124	11,814	44,565	103,295	12,405	6,779	7,849	12,059	184,175	7,820	1,113	1,567,882
1897	952,486	6,601	12,431	13,539	49,341	139,728	20,351	12,910	6,740	17,248	86,838	13,711	990	1,331,924
1898	1,367,570	5,432	25,021	12,703	67,160	153,147	19,468	26,530	11,424	22,670	173,250	8,968	1,290	1,893,343
1899	1,416,791	9,248	8,464	26,613	89,794	211,684	34,384	49,472	6,775	27,853	224,278	16,382	1,269	2,121,738
1900	1,442,471	14,889	9,563	11,060	116,858	204,731	45,074	67,522	4,543	32,751	333,527	21,174	1,364	2,304,163
1901	1,524,739	20,013	28,134	22,856	177,543	217,593	29,927	60,526	37,208	36,875	433,235	28,890	1,538	2,617,539
1902	1,427,558	15,192	28,819	27,148	192,196	213,961	69,101	46,270	27,181	37,878	379,174	38,277	1,650	2,502,755
1903	1,190,618	11,328	34,690	28,609	92,768	184,494	59,174	50,089	34,514	41,599	381,820	35,928	1,650	2,145,631
1904	1,205,162	16,645	30,965	15,859	81,245	202,197	55,636	30,016	32,969	50,469	238,518	28,254	1,643	1,987,935

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The number of vessels launched during the quarter was 160 for steam, and seventeen for sail. Of the number now building, 282 steamships and 25 sailing vessels will be under the British flag, being intended for the United Kingdom or the Colonies, and represent a tonnage of 834,427. Germany takes three steamships of large size, representing 34,450 tons; Norway nine, representing 33,147 tons; Holland four, representing 19,460 tons; Austria-Hungary five, representing 18,270 tons; and Denmark five, representing 10,185 tons.

Among the vessels now under construction are five, each having a tonnage of over 15,000, and three of from 10,000 to 11,999 tons each. Of war vessels now building, there are eight under construction at the Royal Dockyards and 46 in the yards of private firms. Of the former, four are first-class battleships, having a total displacement of 65,400 tons; three are first-class armored cruisers, having a displacement of 37,950 tons; and one is a second-class protected cruiser, with a displacement of 5,800 tons. Of those in hand at private yards, nine are first-class armored cruisers, eight are scouts, sixteen are torpedo-boat destroyers, and ten are submarine boats. These are all for the British Government; while, in addition, two battleships and one torpedo-boat destroyer are being constructed by private firms for foreign countries. This makes a total of 54 war vessels in all, with a total displacement of 282,810 tons. Two of the battleships are under construction at Portsmouth, one at Chatham, and one at Devonport. One of the first-class armored cruisers is being built at Chatham, and two at Pembroke; while one second-class protected cruiser is in hand at Devonport. The work at the private yards is divided between Barrow, Blackwall, Clydebank, Elswick, Govan, Greenock, Birkenhead, Chiswick, Cowes, Hebburn, Jarow and Poplar.

In connection with the above, which is from "Engineering," London, the tables on pages 140–141 taken from Lloyd's Register are of interest.

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TRIPLE-SCREW PROPULSION.

By REAR ADMIRAL GEORGE W. MELVILLE, U. S. N., Retired, in "Marine Engineering".

The lessons of the war in the far east are more than interesting to the engineering profession, particularly so to that which relates to ship construction and marine engineering.

The order of our President is explicit, that no officer of the Army or Navy of the United States shall enter into any discussion of the relative merits of either nation or course of action of either their fleets or armies. It is not only just that the President should issue such an order, but that the order should be implicitly obeyed. But this does not bar us from learning lessons for ourselves that may be applied to the benefit of our Navy in our new constructions. The first lesson we have learned is the necessity for a high-powered fleet of fast ships, either battleships or armored cruisers, with great coal endurance and perfect protection to the propelling power and steering gear of this class of ships.

The lesson of the loss of the *Rurick*, the Russian armor-clad ship, the consequent delay and scattering of the Russian fleet at that time, the forced return of some of the ships of that fleet to Port Arthur, and subsequent abortive attempt to join Russia's Vladivostock and Port Arthur fleets into one strong fighting unit, may be traced to the fact that the *Rurick's* steering gear or rudder and part of her propelling power were disabled at the same time.

As the report comes to us, it is said that a shell from one of the Japanese ships exploded under the stern of the *Rurick*, disabling her rudder or steering gear, and at the same time disabling one of her twin screws. Now, had this ship been a three-screw ship, as nearly all of Russia's modern ships are now fitted, as well as those of Germany and France, it can readily be seen that had her center screw and one of her wing screws remained intact, the ship could have proceeded with at least two-thirds power, and in an emergency even greater power could have been crowded through the two remaining engines and screws, so that not only the *Rurick* might have escaped,

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but possibly the whole fleet. As it was, because of the delay of more than one hour, when the Russian fleet was trying to blanket and protect the damaged ship, the time thus lost, if skillfully used, even in a running fight, might have saved the fleet to Russia. Whether this reasoning would apply in all cases and under all conditions is a question that is is impossible to decide off-hand, for so many contingencies arise in battle that no fixed rule can be made to fit them all.

Accompanying this prelude to my subject, I take pleasure in stating the following reasons why the three-screw system should be applied to naval vessels of more than 3,000 tons, bearing in mind that it is not a new or novel system, but that it is in almost universal use in the navies of Russia, Germany and France; Great Britain and the United States alone neglecting this great military advantage in time of war and the greatest source of economy in coal in time of peace or on blockade.

Not only for men-of-war alone, but for the high-powered transatlantic liners, the three-screw system would prove most advantageous in case of accident to either engine or rudder, and as the increase of engineer force in engine or fire rooms amounts to but six per cent. of the number of men, it is a small amount to pay for the increased security in mail-passenger service.

In the United States protected cruisers *Columbia* and *Minneapolis* the additional force employed for each ship in the engine room, for oilers, was but nine men, or three on each watch for three watches when the ship was under way. Now that we are having numerous three-screwed turbine ships projected and being built, and many more projected with four shafts and propellers, three-screwed ships will cease to be a novelty, and the traveling public on our high-powered transatlantic liners will demand the three-screw system as they now do the twin-screw, and avoid a single-screw ship. After years of service, the machinery of the *Columbia* and *Minneapolis* is reported in as good condition as when built, and the fleet engineer of the training squadron reports to the Navy

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Department that the ships are able to make the speed they did on their trial trips, which speaks well for their design, builders, and the care and attention they have had from the engineer force since they were first commissioned. In fact, the adoption of the three-screw system for ship propulsion will act as rudder and propeller if rudder or steering gear and any one of the three screws or engines is disabled.

The following are a few of the reasons in favor of the three-screw system. I have also stated some of the disadvantages that might arise from the system:

The triple screw is no novelty in naval construction. It has been used in Russia, France and Germany. Russia possesses eleven naval ships fitted with triple-screw engines, and these ships are of late design. France has ten battleships and nineteen cruisers of this type, and the latest armored cruiser launched, *Dupetit-Thouars*, is fitted with triple screws. In the case of Germany, the decision has been practically reached, after competitive trial of the twin and triple-screw systems, to adopt the triple-screw design exclusively in all future warships of the home fleet. She has already nineteen such vessels.

England alone, of the strong naval powers, continues to design twin screws for her most powerful vessels. In doing this she is only pursuing that same conservative policy which makes her refuse to drive her turrets electrically, and which impels her to shun superimposed gun turrets. If this conservatism upon the part of England, as respects the question of triple screws, is based upon sound engineering design, then what reason have we for maintaining that the British Admiralty is not equally justified in its decision to adhere to the hydraulic working of turrets, and in its determination to reject, for all classes of ships, the superimposed arrangement for heavy guns. Concerning the conservatism of the British Admiralty "Marine Rundschau," a leading German periodical, in its issue of March, 1901, says:

"It must be pointed out how inordinately conservative they (the English) are for holding on to old methods, and how they object to anything new in their navy. It was only after the

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French had built ironclads, and had successfully tried them, that the English decided to change their 'wooden walls' to iron ones; and they were many years behind other navies in adopting breech-loading rifles."

In matters pertaining to the naval personnel, it has been the practice of Great Britain to follow rather than to lead our service. In matters pertaining to the material she has always assumed the conservative attitude, maintaining the position that as the greatest sea power she could rest well content with existing conditions, and that such construction as submarine and commerce destroyers are the expedients of weaker powers. Therefore, with respect to the best distribution of motive power, as well as with respect to the question of the arrangement and distribution of battery, the British Admiralty is regarded rather as observers than opponents of American precedents.

In adopting the triple-screw system there will not be introduced either any innovation or discredited design of motive power. The demand for this character of installation is made rather to prevent us falling behind our competitors in naval construction than in the desire to lead our rivals. The practical reasons for thus distributing the power in armored vessels may be embraced in three distinctive groups.

ECONOMIC ADVANTAGES.

Where judgment has been exercised in utilizing the power, marked economy has been secured by the use of triple screws. The economy of this installation is so well recognized by the Germans that the matter is no longer in dispute. It would be remarkable if there were not economic advantages. There must be a saving of coal where the distribution of power is of such nature that only the amount of machinery need be put in operation which can be run at economical speed.

The official trials of the *Columbia* and *Minneapolis*, when compared with those of the *New York* and *Olympia*, prove conclusively that there was a decided economy of propulsion. Probably no data were ever obtained with more care than in the official trials of the four ships named. There was an ample

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corps of trained observers. The system of observations was the result of experience after many official trials. The indicators had been most carefully calibrated by the same experts in all cases, and the apparent pressures were corrected for the errors of the instrument. There can be absolutely no doubt of the accuracy of the data.

Some of the special economical advantages are the following:

(1) In time of war, or emergency, the ordinary cruising can be maintained by the side engines, while the center engine is run at just sufficient speed to clear its propeller. Where there is a necessity for an economical expenditure of coal and an equal necessity for being in readiness to move quickly, the center engine can be run slowly while the side engines can just turn over. In either of the above cases the load on the engine that is just turning over is only that due to friction, and therefore the coal expenditure is exceedingly small. Such operation of triple-screw engines was actually done on the *Minneapolis* during the late war. When operating the engines in this manner it is only necessary to open all the throttles to secure a maximum speed. Such a condition or readiness, at a slight cost of fuel, can only be secured with triple screws.

(2) At full power the economy will be equal to or greater than with twin screws. At reduced speeds the economy will be much greater, thus increasing the steaming radius and keeping the vessel longer on blockade or other important duty.

(3) The economy is secured by more efficient propulsion. It is also attained by the more efficient use of steam in the engine, due to decreased condensation following the increased piston speed.

(4) The economy of the triple-screw system has been shown by actual experiments and experiences in our service. These results have been verified by the experience of the French and the Germans. Russia has not conducted comparative tests, but the general belief exists in that service that the triple-screw system is the most economical.

The economic superiority of the triple-screw design is now practically recognized by every nation that has possessed

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ships of the two types. Such judgment has been founded upon experience and experiments. A design of motive power which secured a reduction in the consumption of coal is of great military advantage. "Coal, or the want of it, is the life or death of a fleet." Its consumption must be reduced on the modern war ship, and the great trend of marine engineering research is in the direction of such economy. To continue building twin-screw engines in high-powered naval vessels is to be one engine behind in the race for naval supremacy.

STRUCTURAL ADVANTAGES.

In addition to the economic advantages resulting from the design, there are structural reasons why the system should be adopted. The following are a few of the resulting advantages:

(1) The engines can be made lighter, cheaper and simpler.

(2) The subdivision of watertight compartments is more efficiently secured.

(3) The nearer the screw to the center of the vessel the more efficient it is. The screw need not extend so far beyond the lines of the hull. The screw is less liable to disablement from ramming any floating obstructions.

(4) Shorter lengths of shafting can be used. The use of struts can be obviated under certain conditions.

(5) Smaller engine parts can be used. This is a marked military advantage, since it will not only reduce the cost of repairs, but permit the repairs to be made more rapidly as well as by a greater number of engine repair shops.

(6) The vibration is decreased. This advantage is of great importance, since it not only adds to the endurance and efficiency of the machinery, but contributes to the comfort of the crew, and passengers, if a passenger ship.

(7) Triple screws permit the installation of additional auxiliaries. The auxiliaries are the weak links in a modern battleship, and by the disablement of a single one a set of propelling engine can be thrown out of use. Where three sets of auxiliaries are installed, it would be but seldom that two sets of engines could not be kept in operation. In fact, two

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auxiliaries of any one description could be so forced that it might not be necessary to stop working any of the propelling engines if an auxiliary was disabled.

(8) More latitude is allowable in the design of machinery. Smaller and more efficient propellers can be designed, the piston speed increased, the foundations made lighter, the working parts smaller, and all permit greater ease of repairs.

Structural reasons first impelled the marine engine designer to suggest the triple-screw engines. The advantages were so apparent that the installation was secured. The resulting structural and engineering benefits were very great, and yet it is but fair to recount the disadvantages which the system entails.

The following are the principal structural disadvantages:

(a) Greater number of parts. In all high-powered naval vessels fitted with twin screws there are two sets of engines coupled on to each shaft. Six cylinders with their attendant working parts are required for each set of machinery, so that as a matter of fact there are more working parts with the twin-screw system of the same power.

(b) Greater number of valves and pipes; but these fittings are smaller, and therefore more easily overhauled and repaired.

(c) Greater number of auxiliaries. In the cross security of being able to use any auxiliary on any propelling engine this ought to be a marked advantage.

(d) Increased engine-room force. The total increase in the fire and engine-room force is about six per cent.

(e) Intricacy of piping. Additional piping, of course, is required, but no man competent to run high-powered machinery would regard the additional lengths of piping as producing complication, and in all cases the piping is smaller in proportion to the unit used.

(f) The engine space requires more room lengthwise. Space is utilized in the afterpart of the vessel, however, that could be only used as store rooms for spare gear.

The structural advantages so far exceed the disadvantages that from an engineering standpoint there can be no justifica-

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tion henceforth for the use of twin screws in large-powered vessels. It must be understood that by the adoption of the triple-screw system we use three sets of engines instead of four. The *New York* and the *Brooklyn* have two sets of engines coupled to each of the twin-screw shafts for economy in fuel at low cruising speed. The experience of the *Brooklyn* at Santiago shows that where two sets of engines are coupled to the same shaft, the vessel would practically go out of action if the attempt were made to couple the shafts in times of emergency. These are sufficient structural reasons alone to justify the adoption of the system.

TACTICAL ADVANTAGES.

In addition to the economical and structural reasons for installing triple screws there are tactical advantages to be secured. When the German Admiralty began giving consideration to this subject her experts did not look for other than economical and structural advantages. In fact, it was believed that investigation would show that tactical reasons would prevent the further use of triple screws in German warships. With the thoroughness that characterizes German research, extended experiments were planned for the determination of this question. The *Friedrich III*, fitted with triple screws, was pitted against the twin-screw ships *Brandenburg* and *Sachsen*. All manner of tactical evolutions were attempted. To the surprise of the German experts the *Friedrich III*, fitted with triple screws, showed a marked tactical superiority over her rivals.

These experiments caused increased attention to be given to the tactical reasons for adopting the triple-screw system. It was found that the ships fitted with such installations steered best, and that the art of steering was more easily acquired by the sailors, as the ships took no rank sheer, as is found to be the case with vessels carrying twin or single screws.

As confirmatory of the *Friedrich III-Sachsen* experiments, the following particulars are given: The first squadron of

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German battleships, comprising three vessels of the *Imperial* type—*Emperor William the Great*, *Emperor William the Second*, and *Emperor Barbarossa*—fitted with triple screws, made two long cruises during the past summer. It is reported of them that they could be more easily steered and maneuvered, a new proof of the advantages of the triple screw over the double propelling system of the *Brandenburg* type.

The remarkable fact is now presented of the German experts declaring that it is principally for tactical reasons, and not by reason of the structural and economical advantages, that Germany has settled upon the policy of using triple screws for all large-powered ships in home waters. The following are the principal tactical advantages:

(1) Increased assurance of working the entire machinery. This is of special importance in battle or in time of emergency, when maximum power is required for a short period. At Santiago the *Brooklyn* could not use her maximum power because only the after engines of each shaft could be used, the forward engines being uncoupled.

(2) Decreased liability of imperiling the motive and maneuvering power of the vessel. This advantage arises from being able to install the screws near the center of the vessel, thus decreasing the chances of the propellers being hit by a floating obstruction.

(3) The steering or maneuvering qualities of the ship are not so likely to be interfered with in triple as in double screws, since it is not likely that more than one engine will be disabled, as in the case of the Russian battleship *Rurick*.

(4) A tactical advantage, not hitherto shown, is better control when under stern board. An experiment with the *Columbia* when in a narrow channel went to show that she could readily be steered when going astern by carefully handling the engines and disregarding the helm, the helm being amidships. This we know cannot be done with either single or twin-screw vessels.

(5) A helm angle from 6½ to 10 degrees is required to keep

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a twin screw on her course when only one screw is used. No such angle would be required when one of the three screws is disabled.

(6) Greater safety of machinery due to better distribution of power.

(7) In case the rudder is carried away or disabled the triple-screw ship can be more easily handled, since the wing screws can be used for guiding and the center one for propelling the vessel—again the *Rurick*.

(8) All the tactical advantages secured by twin-screw vessels are maintained; additional ones are attainable.

The *Brandenburg-Sachsen* experiment showed that evolutions could be performed with triple screws that could not be done with twin screws. The March number of the "Marine Rundshau" gives detailed description of buoy experiments which could only be performed by ships fitted with triple screws.

Since the triple-screw design permits smaller engines to be used, there is a tactical and military advantage in such construction. Smaller engines are cheaper, lighter, simpler, more easily managed and more easily overhauled. This may be of great significance, now that we are finding such difficulty in securing trained men for the engine rooms of our warships. In view, therefore, of the economical advantages, as shown upon official trials; of structural advantages, as recognized by marine engineers, designers and builders, and of tactical advantages, as determined by careful experimental investigation and research, there should be no hesitancy in using the system.

The distribution of the motive power is no less important than the distribution and arrangement of the battery. It is just as logical to arrange the battery of the new warships in the same manner as was done a decade ago as it is to continue the custom of installing a twin-screw arrangement of motive power that is not in accord with existing needs.

For the past five years I have continuously and consistently urged the installation of triple screws for high-powered ships. My only regret now is that I was not more insistent in urging

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the adoption of the system. Under several occasions I determined to present a minority report against twin-screw engine propulsion for high-powered ships. Such official protests were not made because the assurance was given that there were grave tactical objections to its introduction, and that data to support the contention would be furnished. With each succeeding year progressive evidence has been adduced that the system is not only in accordance with sound engineering practice, but it supplies a tactical need of the future battleship.

The triple-screw system for high-powered naval vessels is no longer either an innovation or an experiment—it is a military necessity. The continued installation of twin screws in such vessels is not only against progressive engine design, but fails to meet the military and tactical requirements of the future. If such practice continues our naval prestige will be lessened. To defer adopting triple screws is marked retrogression. The distribution of power by triple screws is simply keeping pace with the best naval construction of Europe. Any action of a contrary nature is simply discrediting the ships in process of design. It is, therefore, imperative that this system be adopted unless the facts herein set forth can be controverted.

SUPREMACY OF THE BATTLESHIP.

That the impression which has gained credence in some quarters to the effect that the efficiency of the battleship has been discounted by the operations in the far Eastern war may be corrected, the Navy Department has issued the following official statement commenting upon the naval operations of the Russian and Japanese fleets and calling attention to the fact that absolute supremacy of the battleship has throughout the war been again and again demonstrated. The statement was prepared by a member of the General Board who is well known as a tactical and strategical expert and represents the views of the Board and of the Secretary of the Navy. It follows:

"The present war in the far East has in its many-sidedness

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given pause to many whose vocation and tastes lead them to ponder questions of military policy and strategy. The conduct of the Japanese armies has won world-wide admiration, equalled only perhaps by that aroused by the heroic resistance of the Russian garrison at Port Arthur.

"A still greater surprise, perhaps, has been caused by the superb condition and efficiency of the Japanese fleet, ascendancy upon the seas being supposedly a less natural development of their old-time war spirit than a fighting supremacy on shore. Every type of war vessel has its place in the navy of that country, and it is not too much to say that each has been used in the manner and for the purpose to which it is best adapted and for which it was intended; and perhaps no more useful book is opened to the modern tactician than a thorough and logical study of the employment of the different weapons. In such a study sound conclusions may differ surprisingly from hasty conceptions.

"To neutralize something powerful and costly by means of an inferior and cheap agent has always been an attractive theme; but seldom, if ever, have the ends attained justified the brilliant expectations of those who have allowed their imaginations to run riot. In 1805, William Pitt, upon examining the plans of Fulton's torpedo boat, remarked that if that mode of fighting were adopted it would be the end of military navies; and yet a hundred years have passed and the battleship still stands supreme. During these generations the French, despairing of opposing equal force to the British fleets, had adopted the policy of employing swift cruisers for commerce destroying, and they actually did capture a great many merchant ships of their hereditary enemy; but the injury was confined to the feelings of the many and the pockets of the few; supremacy on the sea and mastery of the situation remained with the nation whose naval line-of-battle stood unbroken. In the early days of the modern development of the torpedo boat, the fascinating idea again came to the front of using those vicious little craft to annihilate fleets; the imaginary contest was likened to that of the microbe attacking the

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elephant; but the revulsion has not come; experience gradually caused it to be seen that the laws of nature remain the same, and that a small and frail cockleshell cannot maintain either speed or even existence at sea in the same manner as a heavy ship; the results of the present war in the East appear to demonstrate that not only is the availability of a torpedo boat at the front dependent upon having a supporting ship and convenient base and refuge, but that prospect of success is greatly enhanced by the presence of a containing force to restrain the enemy and keep him in a situation favorable to the attack. It has been suggested that even the dormant mine has in this war been so successful as to almost enter the category of *offensive* weapons and thus still farther discountenance the battleship; in its important, though somewhat accidental results, the mine may be likened to a small chemical pellet which will surely prostrate the strongest man in his carelessness, but which can surely not do a man's work in the world.

"Ships must not touch mines, and must protect themselves against the torpedo, but their power in bringing war to a successful issue never has been stayed, except by a greater force of a similar nature. And an examination of the results of the various modes of attack brings out the additional evidence of this war as indicating the probable immutability of that governing principle.

"In the brusque inauguration of hostilities by the Japanese torpedo-boat flotilla at Port Arthur, during the night of February 9, they did undoubtedly deliver a telling blow by disabling three Russian battleships. The temporary crippling of that force affected the conduct of operations, not only during the beginning, but throughout the entire first phase of the war; the Japanese, having by that stroke won the mastery of the sea, were able to land troops at the Yalu and later at Niuchwang, instead at the far south end of the Korean Peninsula. This mastery was due to their having the preponderating force of battleships, which alone enabled them to blockade the Russian fleet. No more eloquent tribute to the pre-eminent value

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of battleships is to be found than this one circumstance, that the small margin of superiority thus obtained saved the army hundreds of miles of precarious and weary land transport of men and supplies, enabling them to strike swiftly, hard and continuously upon a weaker enemy before he could be reinforced.

"On the other hand if, under normal conditions, battleships can be destroyed or seriously damaged by unsupported torpedo boats their value is seriously compromised. This, however, does not appear to be the case. The action of February 9 was a surprise far excelling the usual elements of a surprise attack; the Russian fleet, fourteen vessels in all, were anchored close together in the outer funnel-shaped roadstead, without suspicion of impending hostilities, not a patrol boat being out, and the crews peacefully sleeping in their usual berths; three Japanese destroyer flotillas (twelve vessels) approached with lights burning, and being mistaken (as intended) for Russian boats returning, were allowed to approach close to the huddled fleet; then, with a smooth sea, at close range, hardly disturbed by a single shot in return or flash of searchlight, with all the conditions of easy target practice, not less than twenty-four torpedoes (and probably more) were discharged, of which three took effect. The results were undoubtedly momentous; but it would be a hardy logician who could deduce from that incident the comforting thought that the little torpedo craft has displaced the battleship or even materially diminished its prestige. As is well known to all who have followed in the press the procession of events, Admiral Togo's fleet left Japan at the time the message was sent recalling the Minister at St. Petersburg, established a naval base in the islands neighboring to Port Arthur, and thenceforth, with his battleships and large cruisers, invested that port and supported his small craft in their constant attacks; protected from torpedo attack by his own torpedo flotillas, he blockaded the Russian fleet and immobilized their battleships. Although the Russian destroyers in Port Arthur numbered

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twenty-four, or eight more than all in the U. S. Navy, not one has scored a hit so far during the war.

"Scores of other torpedo attacks have been delivered by the Japanese, but nearly all met with complete lack of success, although made, as usual, under the cover of darkness and against an enemy at anchor and whose position was known. Opon one occasion, after a soul-stirring night, that of June 23, the Russian ships being in the outer roadstead, twelve torpedoes were picked up in the morning, having missed their marks and having their mechanism apparently set to float at the end of the run (instead of to sink, as is customary), in the evident hope that they might find a target while drifting about. How many others were sped during that strenuous night, and sank or drifted away in the tide, will probably never be known. In a few other cases hits were scored which laid the victims up for a few days, after which they returned to service. But the fact remains that not a Russian has been sunk by a torpedo until just recently, when the *Sevastopol* succumbed to the incessant attacks, she being 'contained' by the Japanese battle fleet and compelled to lie helplessly anchored ouside of the harbor, entirely exposed, and with many of her guns landed to aid in the shore defenses. The circumstances were indeed peculiar. The fleet in Port Arthur had been destroyed by gun fire, the Japanese having occupied a hill which allowed them to deliver a direct fire on it with siege and naval guns. The *Sevastopol* was moved to the outer roadstead, and, hemmed in by Togo's battleships, anchored under the Tiger's Tail forts, protecting herself by a boom defense and nets of other vessels lost. This defense withstood continuous night attacks by Japanese destroyers and torpedo boats for ten days, when she was finally struck by three torpedoes, and is now beached on the rocks under Golden Hill forts. She was a stationary target day and night for two weeks, and it is not known how many attacks were delivered against her, but a probably low estimate of the number of torpedoes fired would be 150. The Japanese suffered severely in the attacks, and lost two destroyers and some of the smaller torpedo boats. The weather, while severe

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upon the boats' crews, was evidently favorable for torpedo work, as the reports refer to snow, rain, sleet and the elements generally being bad.

"Day attacks by torpedo-boats have never been regarded as worth attempting. Several incidents of this war have been illustrative of their impotence in daylight. The first was when the *Variag* came out of Chemulpo to face the Japanese squadron; not a single torpedo-boat was permitted to enter the field of action. Again, during and following the fleet action of August 10, although there were present eight Russian destroyers and at least fifteen Japanese destroyers and many torpedo-boats, not a torpedo reached its mark, if, indeed, any were discharged; none dared to approach within torpedo range by daylight. The same is true of the action of August 14, between the Vladivostok armored cruisers and Admiral Kamimura's fleet; the ill-fated *Rurik* was crushed by gun fire alone, and the torpedo-boats effected nothing against the *Rossia* and *Gromoboi*, which escaped to Vladivostok, although badly damaged by gun fire. So far as actions on the open sea are concerned, it may be said that in none have the results been influenced in any way by the presence of the torpedo-boat or the torpedo-boat destroyer.

"Somewhat to the chagrin of students of the game of naval warfare, this war has afforded no field for the discussion of cruisers. The armored cruisers of the two fleets have simply been mutually pitted one against the other; 'protected' cruisers, whose whole sphere in military operations has of late years been admitted to be restricted to extended sea-scouting, have been barred from that service by the localized character of the operations, the smaller craft, with their wireless telegraph systems, being on constant sentry duty off Port Arthur and notifying Admiral Togo of any threatened sortie. Vessels of all types were present in both fleet actions, but the attention of both sides appears to have been devoted exclusively to crippling each other's battleships. Nor have there been any individual actions between unarmored ships, other than the destruction of the gallant little *Novik* by the *Chitose* and *Tsus-*

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hima. The *Variag*, upon issuing from Chemulpo, was, of course, immediately crushed by the overwhelming force confronting her. The other Russian cruisers were of occasional assistance in driving off the enemy's torpedo boats or rescuing their own; but rarely has the name of any such vessel caused occasion for mention in either official or press reports.

"The overshadowing importance of the battleship is nowhere more appreciated than in Japan. Had they had at the outset twelve, or even ten, battleships instead of six, the Russian fleet would have been destroyed on August 10—the supreme incentive to holding out at Port Arthur being thus removed, that fortress would probably have fallen, the war correspondingly advanced and thousands of lives and millions of treasure saved. Threatened now by the Russian second Pacific fleet, it is safe to say that they would gladly recover at several times their cost the *Hatsuse* and the *Yashima* (if so be it that the uncontradicted reports of the latter's loss be true); for well do they know that the loss of the command of the sea would mean the end of the campaign in Manchuria. Likewise gladly would the Russians buy back the *Petropavlosk*, sunk in the same manner as the two Japanese—by mines skilfully planted by the enemy.

"In summing up the lessons of the war one cannot do so more accurately or felicitously than was done by Lord Selborne, First Lord of the British Admiralty, in the following words:

"'The lessons from the war in the Far East are the importance of the personnel, the necessity for having a margin of strength, and the fact that without battleships no power can hold or win command of the sea.'"—"Army and Navy Journal."

THE BEST ECONOMY OF THE PISTON STEAM ENGINE AT THE ADVENT OF THE STEAM TURBINE.*

By Professor J. E. Denton, M. E.

When James Watt retired from business, at the beginning of the nineteenth century, he had declared his belief, based on

* A paper presented before the Mechanical section of the International Congress of Arts and Sciences at St. Louis, September 23, 1904.

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theoretical calculations, that the water consumption of a steam engine per horsepower decreased with the increase of the ratio of expansion until the pressure of release should be equal to the exhaust pressure. But, although Watt invented a valve gear with which to test his theory, he never worked steam expansively to any practical degree in any of the engines built by him, because the demand for them simply as multipliers of the power available from other prime movers of that day was at too great a premium to make it necessary for the firm of Boulton & Watt to improve the economy of their engine by the application of the principle of expansion.

As builders of steam engines multiplied, however, economy became the ruling factor of competition, and the principle of expansion was applied in the pumping engines of Cornwall, with increasing economy, up to the greatest ratio of expansion consistent with commercial limits of size of cylinder per horsepower. Hence, when in 1840 one of the leading Cornish pumps was reliably tested for water consumption at ratios of expansion varying from 1.5 to 3.5, with results showing from 24 to 16.5 pounds of steam per horsepower hour, and it was found that these figures agreed with Watt's theory, belief in the latter was practically universal, and expansion was relied upon to yield similar results, in all important engines of land and sea, up to 1855.

The engineers of this time failed to realize, however, that the Cornish pump, with its single action and equilibrium exhaust, never exposed the working end of its cylinder to the temperature of the condenser, so that the range of temperature to which the admission end of the cylinder was exposed was only a fraction of that undergone by the steam, a fact which realized the conditions of action of a compound engine. They also failed to realize that the steam was superheated during admission by excessive wiredrawing from the boiler pressure of 45 pounds to 18 pounds absolute, and further, that a live-steam jacket heated the cylinder walls to initial temperature during a very slow indoor stroke, while admission occurred during the very quick outdoor stroke.

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These facts made it possible for the Cornish pump to act without cylinder condensation—a phenomenon which Watt's theory did not recognize to exist; so that it was consistent that his theory should completely account for the performance of this engine as determined by experiment. But the ordinary double-acting non-jacketed engines of the marine and stationary practice of that day did not realize these passive cylinder-wall actions of the Cornish pump. Consequently it became gradually evident toward 1860 that the steam engines of the world were failing to realize the results due to Watt's theory. This conclusion was deduced from service records of coal-consumption tests, as from 1840 to 1860 no impressive water-consumption tests were made.

At the latter date, however, Isherwood's elaborate and thorough tests on the United States steamer *Michigan* appeared, showing a limit to the gain in steam economy by increasing ratios of expansion through the existence of some action of the steam unknown in its character but evidently not taken account of by Watt's simple theory of expansion.

Rankine's sagacious analysis of the *Michigan* data located this missing factor of the theory in the powerful waste action of the cylinder walls, constituting, with all of its complicated dependence on range of temperature, cylinder surface, speed, and jacket influence, what we now term the "loss due to cylinder condensation." It was then quickly recognized that to realize the gain due to Watt's theory of expansion, the percentage of loss due to cylinder condensation must be confined within certain limits as the ratio of expansion increased, and that it could be by controlling its main factor (the range of temperature in the cylinder during expansion), through the division of this expansion among successive cylinders.

DEVELOPMENT OF MARINE ECONOMY.

The idea of the subdivision of the range of temperature was first extensively applied in marine practice to compound the simple condensing engine using steam at 60 to 80 pounds boiler pressure with six expansions without either increasing

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the boiler pressure or expansion ratio. No practical gain in economy resulted from this change, as, while the range of temperature of the cylinder was cut in two and the cylinder condensation loss thereby reduced, the gain accruing therefrom was neutralized by the mutilation of card area, due to the discontinuity of the expansion curve and the loss of pressure between cylinders. The water consumption, therefore, remained at about 19.5 pounds per horsepower, which had been afforded by the simple engine.

But the marine boiler-making art had now advanced so as to make 150 pounds steam pressure available. Therefore, the engine was able to use fourteen expansions and secure such an increase in mean effective pressure as would enable a 3-cylinder engine to weigh practically no more per horsepower than did that of the compound system. The marine world, therefore, sought the increased economy due to Watt's theory from fourteen expansions divided among three cylinders, so that the range of temperature in each cylinder was the same as that in the compound system with lower boiler pressure. The cylinder-condensation loss was thereby controlled so that the gain in economy due to the greater ratio of expansion under Watt's law preponderated over the loss due to broken card area. The result was, taking the merchant marine engines as the standard of economy, that the water consumption fell to 15 pounds per horsepower. The triple-expansion engine at sea, therefore, afforded the first real gain in economy over the simple engine of six expansions, which was prevalent when the phenomenon of cylinder condensation was discovered.

Further development of marine boiler construction now supplies the merchant marine with 210 pounds of pressure from Scotch fire-tube boilers. This permits the use of fifteen expansions with 39 pounds of total mean effective pressure, or about 30 per cent. increase over that of the 150 pounds triple-expansion engine; this, together with increased revolutions, by virtue of improvements in lubrication and construction of bearings supplemented by more perfect balancing of moving parts and no greater mutilation of card area than in the triple

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engine, permits the use of four-stage expansion engines whose weight is within satisfactory commercial limits.

With the expansion divided into four stages, in four or more cylinders, the temperature range per cylinder is reduced over that of the triple engine with a resulting reduction of cylinder-condensation loss so as to afford a water consumption of 13.6 pounds per horsepower hour of the main engine, with a vacuum of 25.5 inches. This economy is attained with saturated steam without jackets or reheaters, the latter never having shown sufficient gain on marine engines to compensate for the care necessary to manage them; and it applies to all engines above 5,000 horsepower, the gain in economy due to larger size being indistinguishable by service records.

With an attached air pump, the usual auxiliaries consume steam which increases the water consumption to 14.4 pounds per horsepower of the main engine. The exhaust of these auxiliaries applied to heating the feed water, supplies it to the boilers at fully 200 degrees Fahrenheit. From this temperature Scotch boilers with Howden's system of draft, burning about 21 pounds of the best semi-bituminous coal per square foot of grate per hour, evaporate 11 pounds of actual feed water per pound of coal. Thus for each indicated horsepower of the main engine 1.3 pounds of coal are consumed per hour.

Without an increase in boiler pressure above 210 pounds, further increase in economy can only be obtained with higher ratios of expansion, and this is prohibited because the consequent reduction of mean effective pressure would increase the weight of the engine per horsepower beyond commercially acceptable limits.

With higher boiler pressure greater economy can be obtained from the engine by increasing the expansion up to probably a ratio of 30, but to accomplish this without increase of weight of engine per horsepower would require a working pressure of steam of 300 pounds per square inch. This must be obtained from water-tube boilers, as the Scotch boiler is now at the limit of practicable construction and allowable weight when built for 210 pounds steam pressure.

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Higher economy of engine is obtainable, with present pressures and ratios of expansion by the use of superheated steam, but the application of the latter has not thus far shown a gain in coal economy in trans-Atlantic marine practice. This is doubtless ascribable to the use of imperfectly-designed superheaters; but the increase of cylinder oil which superheating requires makes its successful introduction at sea questionable, on account of the difficulty of protecting the boilers against over-fouling by accumulation of oily deposit.

At the advent of the steam turbine in marine practice, therefore, the degree of economy with which it must compete in marine engines on the largest scale is that of a quadruple-expansion system, expanding saturated steam with 210 pounds pressure fifteen times into a vacuum of 25.5 inches, with an economy of 13.6 pounds of water per horsepower hour of the main engine, which will operate, with its auxiliaries, from Scotch boilers on the Howden system, with a consumption of 1.3 pounds of the best semi-bituminous coal per indicated horsepower hour of the main engine.

If the turbine could operate screw propellers efficiently with the same rotational speeds for a given horsepower that it uses in electric service, the record for economy already established by the turbine shows its ability to easily compete with these conditions of performance of the best marine engine. But such rotational speeds for the turbine are inconsistent with the degree of screw propeller efficiency available from piston engines.

Therefore the possibilities as regards steam economy of the turbine in marine practice are as yet too unsettled to permit of their definite statement within the limits of this paper.

DEVELOPMENT OF STATIONARY-ENGINE ECONOMY.

Stationary engines at the date of the discovery of cylinder condensation were affording the maximum economy in simple condensing cylinders, expanding steam of from 60 to 80 pounds pressure six times, with a vacuum of about 27 inches, which

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afforded a water consumption of about 19 pounds per horsepower hour.

The first noteworthy improvement upon this degree of economy was obtained in the Leavitt pump at Lawrence, Mass., which by expanding 120 pounds steam sixteen times in two cylinders, secured a water consumption of 16.5 pounds per horsepower. This, however, was a compound engine, without a receiver, or what is known as the "Wolff" system, which does not divide the range of temperature due to expansion between the two cylinders, as the high cylinder is in connection with the low during the whole of the latter's expansion period. This fact and the slow speed of 12 revolutions per minute made the initial cylinder condensation 40 per cent., or nearly twice that of the 6-expansion simple-condensing engines, besides which, live steam jackets on both cylinders consumed 10 per cent. of the total steam.

The gain in economy over the simple engine was therefore due entirely to the influence of the high ratio of expansion aided by high initial pressure and so strong a re-evaporative effect that the expansion line was outside of a Mariotte curve to the end of the expansion. The combined effect of these more than offset the negative influence of the high percentage of cylinder condensation. The engine was, however, the most economical of its time (1872), and it is interesting to note that its conditions of steam action were widely in contrast with that of the most economical Cornish pumping engine of 1840, which afforded precisely the same water consumption of 16.5 pounds by expanding 30 pounds steam three and one-half times, with a complete avoidance of loss by cylinder condensation.

The first prominent advance in economy by division of the range of temperature was obtained about 1878 in the Corliss Pawtucket compound pumping engine. This also expanded 120 pounds steam sixteen times in two cylinders, but with a receiver between them; whereby, with the aid of a speed of 50 revolutions and only 5 per cent. jacket consumption, the cylinder-condensation loss was reduced to 25 per cent., and a

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water consumption of only 13.7 pounds resulted. This result led Corliss to introduce similar compound engines into mill practice, and they were then rapidly substituted throughout the world in all large power-using industries for the single 6-expansion condensing engine. All makers of 4-valve drop cut-off engines in both the United States and Europe obtained from them a water consumption of from 13.5 to 14 pounds per horsepower with about 125 pounds steam pressure and about 16 expansions.

The gain from one division of temperature being thus established, triple-expansion engines were quickly introduced for mill and electric service by Corliss and Allis in the United States and Sulzer in Germany to obtain a further gain from two divisions of temperature. None of the triple-expansion engines succeeded, however, in reducing the water consumption below 12.5 pounds per horsepower, which was not a sufficient gain over the compound engine to offset the extra cost and care of the third-expansion stage. Its use was therefore quickly abandoned in mill and electric-light service in favor of the compound engine.

Such a general readoption of the compound type resulted in a variety of cylinder ratios being used as involving possibilities of improved economy over the ratio of 4 to 1 introduced by Corliss. The first radical departure appeared in the Rockwood & Greene system, which was introduced into New England, with a cylinder ratio of about 7 to 1, and clearances of about 2 per cent., instead of the 4 per cent. value previously prevailing. With 150 pounds steam pressure, live-steam jackets on either the heads or barrels of both cylinders, and a live-steam reheating receiver, such engines have afforded 12.45 pounds per horsepower with saturated steam expanded twenty-six times into about 27.5 inches of vacuum. Without the jackets and reheater in use, about 2 per cent. greater water consumption is obtained. The jackets and reheater, when in use, consume only about 8 per cent. of the total steam, with practically 25 per cent. of cylinder condensation under both conditions.

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This performance represents the best economy derived from the compound engine for mill and electric practice up to within a few years ago, and it has largely established the belief that a cylinder ratio of about 7 to 1, with so small a clearance as 2 per cent., is necessary to maximum compound-engine economy. During the last two years, however, the following records have been established:

(1) With an 850-horsepower Rice & Sargent compound Corliss engine running at 120 revolutions per minute, having a 4 to 1 cylinder ratio, clearances of 4 per cent. and 7 per cent., live jackets on cylinder heads and live steam in reheater, Professor Jacobus, of Hoboken, found for 600-horsepower of load, with 150 pounds pressure of saturated steam, 28.6 inches of vacuum, and 33 expansions, 12.1 pounds of water consumption per horsepower—a result produced with a cylinder-condensation loss of 22 per cent., and a jacket consumption of 10.7 per cent. of the total steam consumption.

(2) With a 250-horsepower Belgian Van den Kerchove poppet-valve compound engine, running at 126 revolutions per minute, with 2.97 to 1 cylinder ratio, clearances of 4 per cent., steam-chest jackets on barrels and head, and no reheater, Professor Schröter, of Munich, found 117 horsepower of load, 130 pounds pressure of saturated steam, 27.6 inches of vacuum, and 32 expansions, 11.98 pounds of water per horsepower—a result produced with a cylinder-condensation loss of 23.5 per cent., and a jacket consumption of 7 per cent. of the total steam consumption in the high-cylinder jacket and 7 per cent. in the low jacket.

(3) With the Westinghouse twin low-cylinder, compound combined poppet-valve and Corliss-valve engine, at the New York Edison plant, running 76 revolutions, with 5.8 to 1 cylinder ratio, clearances of 10.5 per cent. and 4 per cent., without jackets or reheater, Messrs. Andrew, Witham and Wells found for the full load of 5,400 horsepower, 185 pounds steam pressure, 27.3 inches of vacuum, and 29 expansions, 11.93 pounds of water per indicated horsepower—a result produced with an initial condensation of about 32 per cent.

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These facts show that the minimum water consumption of the compound mill engine and electric-light engine of the present date, using saturated steam, is not dependent upon any particular cylinder ratio and clearance nor upon any system of jacketing, but that the essential condition is the use of a ratio of expansion of about 30, above which the cylinder-condensation loss is liable to prevail over the influence of the law of expansion. The conclusion appears warranted, therefore, that if this ratio of expansion is secured with any of the current cylinder and clearance ratios, and with any existing system of jackets and reheaters, or without them, a water consumption of 12.4 pounds per horsepower is possible, and that a variation of four-tenths of a pound below or above this figure may occur by the accidental favorable, or unfavorable, jacket and cylinder-wall expenses which are beyond the exact control of the designer.

COMPOUND PISTON ENGINE ECONOMY *VS.* THAT OF STEAM TURBINE.

In order to compare the economy of the piston engine with that of the steam turbine, we must use the water consumption per brake horsepower, since no indicator card is possible from the turbine; and furthermore, we must use the average water consumption for the range of loads to which engines are subject in practice.

In all of the public turbine tests to date, with one exception the output was measured through the electric power of a dynamo whose efficiency is not given for the range of loading employed, so that the average brake horsepower is not known. This exception is the Dean and Main test of a 600-horsepower Westinghouse-Parsons turbine using saturated steam at 150 pounds pressure, and a 28-inch vacuum. We may compare the results of this test with that of the 850-horsepower Rice & Sargent engine, and the 250-horsepower Belgian engine, by assuming that the power absorbed by friction in these engines is 3 per cent. of the indicated load plus the power shown by friction cards taken with the engine unloaded. The latter showed 5 per cent. of the rated power in the case of the Rice

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engine and 8 per cent. in the Belgian engine; so that the assumption regarding the load friction is a reasonable one.

The results are given in the following table, which shows the average water consumption per horsepower, for from 40 per cent. to 125 of full load, to be 14.55 pounds and 14.64 pounds for the two piston engines, and 14.51 pounds for the turbine, or practical equality:

AVERAGE STEAM CONSUMPTION PER BRAKE HORSEPOWER OF PISTON ENGINE *VS.* TURBINE.

Per cent. of full load.	Pounds water per brake H.P. hour.		
	600-H.P. turbine.	850-H.P. Rice engine.	250-H.P. Belgian engine.
125	13.62	13.78	15.10
100	13.91	13.44	14.15
75	14.48	13.66	13.99
41	16.05	17.36	15.31
Average, 85	14.51	14.56	14.64

As an indication of the gain in economy of the turbine with increase of size, we may use the Elberfeld test of a Parsons turbine at the full load of 1,500 electric horsepower, with an allowance of 5 per cent. for attached air pump, and an assumed minimum generator efficiency of 0.95. Its saturated steam consumption, with 150 pounds boiler pressure and 28 inches of vacuum, is then 13.08 pounds per brake horsepowerhour, which shows a gain of 4 per cent. over the 600-horsepower machine.

The 5,400-horsepower Westinghouse compound engine at the New York Edison station, whose friction under full load was cleverly determined at about 4 per cent., affords a brake horsepower consumption of 12.5 pounds for 185 pounds boiler pressure. Crediting the turbine, therefore, with the possible influence of the difference in size and steam pressure, there is again practical equality in economy between it and the piston engine for equal horsepower.

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DEVELOPMENT OF PUMPING-ENGINE ECONOMY.

While the mill and electric practice has developed the compound engine, pumping engines in the United States have been developed in the triple-expansion fly-wheel type to a degree of economy superior to that afforded by any compound mill or electric engine, and, for saturated steam, superior to that of the pumping engines of any other country. This is because their slow speed permits of greater benefit from jackets and reheaters and of less losses from wire drawing and back pressure. These causes, together with the greater subdivision of the range of cylinder, have resulted in records made between 1894 and 1900 of 11.22, 11.26 and 11.05 pounds of saturated steam per indicated horsepower, with 175 pounds steam pressure, and from 25 to 33 expansions, in the cases of the Leavitt, Snow and Allis pumping engines, respectively, the corresponding heat consumptions being by different dispositions of the jacket drainage 204, 208 and 212 thermal units per indicated horsepower minute; while later the Allis pump, with 85 pounds steam pressure, has lowered the record to 10.33 pounds of saturated steam per indicated horsepower, with 196 heat units of consumption per horsepower minute.

The turbine can compete with these pumping engines only through the medium of a multiple-stage centrifugal pump, whose efficiency, taken even at 80 per cent., will probably impose too great a loss of motive power on the turbine system for successful competition with this high class of piston pump, since the friction loss of the latter is only 5 per cent. of the indicated horsepower.

GAIN FROM SUPERHEATING AND FROM THE SULPHUR-DIOXIDE EXHAUST-HEAT ENGINE.

For opening the way to the improvement of the economy of the piston steam engine beyond any degree represented by the results we have thus far presented, we must make our bow exclusively to the Germans. By their revival of the use of superheated steam, after it had been twice abandoned in the history of steam engineering, they have demonstrated the

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practical success of superheating to raise the economy of any engine, whatever its boiler pressure or ratio of expansion, by rendering the steam a more efficient working medium on its entry to the engine. By showing that after the steam has been expanded to its greatest practical limits in piston engines an important amount of power can still be practically abstracted from it in the sulphur-dioxide engine, they have also reduced the waste accompanying its exit from the engine.

Previous attempts to use superheated steam failed because of the lowered efficiency of the boiler through the location of the superheating device in the chimney. Here the superheater was so arranged that the exit temperature of the heated gases had to be in excess of the superheated-steam temperature to such a degree that the boiler's economy was lowered, and also the superheating surface so located and in contact only with steam was rapidly corroded. These defects the Germans have overcome by arranging the superheater so that it receives wet steam at the chimney end of the system, and lies at the middle of the flame current, at the point where the superheated steam is delivered from it to the engine. Thus the hot gases can enter the chimney at no higher temperature than is consistent with the best economy from saturated-steam boilers, and experience thus far shows no impracticable rate of depreciation of the superheater from corrosion.

The difficulty of conveying highly superheated steam from the boiler to the engine without excessive loss of temperature has also been overcome by improved and thicker pipe installation and the use of about half the pipe section usually employed for saturated steam. The generator plant has therefore been perfected so that it does not handicap the value of the superheated steam to the engine, while the latter is adapted to work the steam without lubrication difficulties by the use of poppet valves on at least the initial cylinder.

The maximum of improvement in economy over saturated steam may be briefly summarized in the following extract from an admirable investigation by the most distinguished European authority on engine tests, Professor Schröter, of Munich: The

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engine used was the compound 250-horsepower 13 and 21 by 33-inch Belgian Van den Kerchove construction, previously mentioned, operating with a boiler pressure of 130 pounds, and 27.6 inches of vacuum, with 30.18 inches of barometer. For saturated steam the average water consumption for five divisions of load, between 40 and 125 per cent. of rated load, is 12.45 pounds per English indicated horsepower hour, with 3.6 per cent. of steam condensed in the high jacket at 130 pounds pressure, and 9 per cent. in the low jacket, including the receiver, at 5 pounds pressure. Assuming the jacket drainages applied to heat the feed water from a temperature of 108 degrees in the hot well, 136 degrees Fahrenheit feed temperature would be afforded. Reckoning the expense of heat from this temperature, the consumption is 225 thermal units per indicated horsepower minute.

With steam superheated 224 degrees Fahrenheit above saturation, or at 304 degrees centigrade, the average consumption for the same divisions of load is 10.09 pounds of feed water per horsepower. With one-third of 1 per cent. high-jacket and 3 per cent. low-jacket and receiver consumption, the jacket water would heat the feed water to 115 degrees Fahrenheit at the same hot-well temperature. Reckoning from this temperature, with a specific heat value of 0.6, according to the experiments of Lorenz and of Bach, the heat expense is 209 thermal units per horsepower minute. The average gain by superheating 224 degrees Fahrenheit is therefore 7 per cent. If the feed water were heated to 200 degrees by auxiliaries the heat unit gain would be 8 per cent.

With steam superheated 307 degrees Fahrenheit above saturation, or at 350 degrees centigrade, and the load at about 80 per cent. of rating, the water consumption was 8.99 pounds per English horsepower, with 0.4 of 1 per cent. condensation in the high jacket and 4 per cent. in the low. This affords 114 degrees Fahrenheit temperature, and with a specific heat of say 0.57 (since according to Lorenz's experiments the specific heat reduces with increase of superheat at any given pressure), we have a heat consumption of 192 thermal units per horsepower

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minute. The same load with saturated steam affords 137 degrees Fahrenheit of feed temperature and gives 221 units of heat consumption per horsepower. The gain for 307 degrees superheating over saturated steam for the same horsepower is therefore 13 per cent. from the natural-feed temperature and for 200 degrees feed 14 per cent.

The best performance reported for superheated steam used in the turbine is that of Brown & Boveri Parsons Frankfort 4,000-horsepower machine, which, with 183 pounds boiler pressure above the atmosphere and 190 degrees Fahrenheit superheat, afforded 10.28 pounds per brake horsepower, assuming a generator efficiency of 0.95. Reckoning from the feed temperature of its vacuum of 27.5 inches, and using a specific heat value of 0.65, the heat consumption is 214 thermal units per horsepower minute.

The heat consumption of the 250-horsepower Belgian compound engine per brake horsepower at the highest superheating of 307 degrees Fahrenheit is 220 thermal units. The turbine, therefore, probably holds the record for brake horsepower economy over the piston engine for superheated steam by a margin of about 3 per cent., although had the compound engine been of the same horsepower as the turbine, so that its friction load would be only 8 per cent. of its power instead of the 13 per cent. here allowed, it would have excelled the turbine in brake horsepower economy by a margin of about 2.5 per cent.

THE SULPHUR DIOXIDE ADDENDA.

The most economical piston engines, which, as we have seen, expand steam about thirty times, release their steam at the end of the stroke at about 6 pounds absolute pressure, but they exhaust into a condenser, which, with a vacuum of 28 inches, contains a pressure of about 1 pound absolute per square inch. If the expansion could continue until the pressure of 1 pound was attained before exhaust occurred, considerable more work could be obtained from the steam. This, however, cannot be done in piston engines, for two reasons: First, because the low cylinder would have to be about five

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times greater in volume, which is commercially impracticable; and, second, because the velocity of exit through the largest exhaust ports possible is so great that the frictional resistance of the steam makes the back pressure from 1 to 3 pounds higher than the condenser pressure in the best engines of ordinary piston speed.

All the work due to this extra expansion can be obtained, however, by exhausting the steam from the low cylinder at 6 pounds pressure, against a nest of tubes containing liquid sulphur dioxide, which thereby is boiled to a vapor of about 170 pounds pressure. This can be used to drive the piston of a separate engine or an extra cylinder attached to the main engine, and be thence exhausted into a condenser to be reliquefied. If the sulphur dioxide expands in its cylinder to a pressure corresponding to its temperature of liquefaction, and the latter is that of steam at 1 pound pressure, the work derived from the fluid is the same as the extra work which the steam would have furnished in the main cylinder by continuing to expand to 1 pound pressure and exhausting against 1 pound of constant back pressure. The greatest vacuum required in this system does not need to be more than 20 inches, which is very easily produced and maintained, as compared with a vacuum of 28 inches, corresponding to 1 pound of absolute pressure.

Professor Josse, of Berlin, has perfected this sulphur dioxide system of improvement, and reliable tests have shown that if cooling water of 65 degrees Fahrenheit is available, and to the extent of about twice the quantity usually employed for condensing steam under 28 inches of vacuum, a sulphur-dioxide cylinder of about half the size of the high-pressure cylinder of a compound engine will do sufficient work to improve the best economy of such engines at least 15 per cent. The steam turbine expands its steam to the pressure of its exhaust chamber, and as unlimited escape ports can be provided from this chamber to a condenser, it follows that the turbine can practically expand its steam to the pressure of the condenser. Therefore a steam turbine attached to a piston engine to op-

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erate with the latter's exhaust should effect the same saving as the sulphur dioxide cylinder, and turbines are being applied for this purpose at present by the makers of the Rateau type.

The turbine, therefore, even at this early period of its history, is already a formidable competitor to the piston steam engine in its most economical degree, and that it is so is mainly due to the fact that with no more waste in its steam action from internal friction than is due to the cylinder-wall wastes of the piston steam engine which are absent in the turbine, it more completely realizes the expansive principle enunciated in the infancy of a steam history as the fundamental factor of economy, by its sagacious founder, the immortal James Watt.—"Stevens' Institute Indicator."

FOAMING OF THE WATER IN A STEAM BOILER AND ITS EFFECT.

By Charles H. Haswell, C. M. and N. E.

There is not an element in the effective and economical operation of a steam boiler and attached engine, whether in land or marine service, and especially in the latter, that is pregnant with more results and casualties than the foaming, or "priming" of it (the latter a foreign term neither expressive nor derivative), inasmuch as it involves not only the wasted expenditures of heated water, by its flowing with the steam of operation into the cylinder of an engine, but, when its volume is in excess of that of the clearance space between the piston and the head or bottom of the cylinder, it involves the disruption of one of them. Upon the resulting arrest of the engine and consequent flow of steam to it, the foaming water, except when violently agitated by the motions of the vessel in which it is operated, subsides, and if its surface is below the upper surface of the furnace tubes or flues, as the case may be, it soon renders them incandescent, and as the result a collapse may ensue.

Before essaying to submit a remedy, or even an amelioration of this objectionable and destructive operation of foaming, it

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is proper to consider its origin. Ordinarily, it is the result both of an insufficient height and volume of steam space above the water line and the *per saltum* flow of steam to the engine, consequent upon the periodic operation of the steam valves, which involves such undulations of the surface of the water that foaming ensues. It also occurs when the area of the surface of the water is less than that of the area over the crowns of the furnace.

With boilers, where foaming exists, various expedients are resorted to, either to obtain additional volume of steam space by the addition of a cylindrical vessel, arranged as an integral of the boiler or attached to it, termed the drum, or by concentric cylinders set vertically at the termination of the flues or tubes, and rising to an elevation so far above the water level as to give space for the foaming water to subside before reaching the opening of the steam pipe. Upon this construction the smoke pipe is set, and the hot air from the furnace rising through the inner cylinder or chimney superheats the steam within the ring to an effective and economical degree. Another method is to extend the steam pipe into the boiler and attach it to the under side of the crown of the shell. The pipe is perforated with numerous small openings, and the steam, instead of flowing in a mass to the opening of the steam pipe, flows in minute currents from an extended surface, and foaming is materially arrested. This construction is termed a dry pipe.

Foaming renders it difficult, furthermore, to readily ascertain the actual height of the water in a boiler; and as the removal or correction of this difficulty is imperative, various devices, in addition to the ordinary gauge cocks of manual operation, areused, such as a glass gauge attached to the front of a boiler and connected to it, both above and below the upper surface of the flues or tubes, by small connections, which arrest and reduce the foaming to a degree that shows the actual height of the water. Sometimes a metallic cylindrical vessel with a glass face is used, connected in like manner.

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The loss in the cylinders from the superhydrated steam due to foaming water is a material element in the question of the economy of a steam engine. The conditions, results and constructious here detailed are well known, and they are introduced, not only as a record of the construction of the period, but to give value to the following illustration of a very common and effective cause of foaming in all boilers, and how it can be materially prevented.

In the operation of the furnace of a boiler, the heated water contiguous to the outer sides of it rises in vertical currents to the water line, and in a boiler where the water line is less in width than the extreme width of the furnaces, the ascending of currents of water in the condition of ebullition will be concentrated at the water line between the sides of the boiler and furnace; and hence a concentrated ebullition of the surface of the water or foaming much in excess of that which arises from the ordinary want of steam space will ensue, which in small marine vessels, where the height under the deck is insufficient to admit of a drum, chimney or like construction, is irremediable.—"Scientific American."

TRIALS OF THE TROOPSHIP *DUFFERIN*.

The troopship *Dufferin*, built and engined by Messrs. Vickers Sons & Maxim, Limited, at their Naval Construction Works, Barrow-in-Furness, for the Royal Indian Marine, completed on the 23d instant her official steam trials, and will leave in about ten days for Bombay. The prescribed trials were six runs over the measured mile at 18½ knots, with six hours' continuous steaming at full speed. These tests took place on the measured mile at Skelmorlie, on the Clyde, the vessel being loaded to her full displacement of 7,340 tons, the draught being 19 feet. A mean of mean speed of 18.897 knots was attained with the engines indicating 9,975 indicated horsepower. The six runs showed remarkable uniformity, as is indicated by the following details:

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.....	Mean revolutions.	Mean indicated horsepower.	Time.		Speed.
			min.	*sec.*	*knots.*
First run..	117	9,421	3	12	18.75
Second run	117.3	9,935	3	11¼	18.83
Third run	118.6	10,365	3	9⅔	19.006
Fourth run	118.4	10,135	3	10½	18.898
Fifth run	117.8	10,167	3	10⅜	19
Sixth run	117.6	9,827	3	10¾	18.872

On none of the runs was the speed less than that contracted for, while at no time did there seem any reason why 19 knots should not be maintained for an indefinite period. The mean results, so far as engine performance, are given in the first column of the appended table, while the averages of the hourly readings on the six-hours' test are set out in the second column. It will be noted that the mean power was 9,881 indicated horsepower, and this was maintained very easily and steadily, a characteristic of the design of engines being their ample bearing surfaces.

FULL-SPEED TRIALS OF R. I. M. S. *DUFFERIN*.

.....	Mean of six runs over measured mile.		Mean of six hours' sea trial.	
	Stbd.	Port.	Stbd.	Port.
Mean steam in boilers	177.3	177.5	179	179
at engines	175.6	175.7	178	178
air pressure	2	2	2	2
cut-off in M.P. cylinder	70.8 p. c.	70.4 p. c.	71.4 p. c.	70.8 p. c.
vacuum	26	25.3	26	26
revolutions	117.5	118.1	115.2	116.3
pressure in cylinders, high	77.48	75.34	79.8	78
intermediate,	32.4	32.5	32.6	33.6
low	14.8	14.64	14.8	13.5
Indicated horsepower, high	1,560	1,526	1,575	1,554
intermediate	1,604	1,614	1,568	1,645
low	1,819	1,852	1,849	1,690
Total indicated horsepower	4,983	4,992	4,992	4,889
Gross total indicated horsepower	9,975		9,881	

We described the ship in connection with the launch, but here some reference may be made to the special fittings since

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added to the ship, which, by the way, was designed by Sir Edward Reed, K. C. B., the Consulting Naval Architect for the Royal Indian Marine. Built primarily as a home trooper, the *Dufferin* is fitted and adapted in every way for the transport of native troops as well as European and for ordinary trooping service in India. It is a satisfaction to note that in her everything has been done to insure the comfort of "Tommy Atkins" while afloat. Although 100 feet longer than the *Warren Hastings*, built about twelve years ago by the same firm, the number of European troops carried is the same—viz: 1,100—the additional size allowing for the increased and improved accommodation now demanded for all ranks. Steam-heating pipes are fitted throughout the living spaces below the spar-deck, and electric radiators in the officers' cabins, smoke room, &c., above this deck.

The ventilation of a ship of this class presents very considerable difficulties, more especially as it is imperative to keep the decks clear of ventilating cowls and other obstructions for the massing and drilling of troops. This consideration has led to the adoption of artificial ventilation by means of electric fans throughout the troop spaces, hospitals, stores, &c., with the result that the spar deck on each side is absolutely free from obstructions of any sort for a length of over 250 feet. A special feature is also the fitting of square ports on the sides of the hull throughout the saloons and cabins on the main deck, and throughout the troops' spaces, hospitals and women's quarters on the troop deck. These square ports are fitted with sliding glass sashes, and along the troop deck with sliding jalousie blinds as well. Special arrangements have been made for opening and closing the troop deck ports from inside the ship. The panelling in the saloons, music room, smoke room, &c., is of hard wood throughout, and the cabin fittings generally, while being of quiet and good appearance, are strong and durable.

The decks and other woodwork of the ship are, with few exceptions, of Burmah teak, light wood being unsuitable, for many reasons, in the Indian climate. The sanitary arrange-

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ments are very complete, and marble baths are fitted for the ship's officers and first-class passengers. A large hot-water tank supplies these and other baths with hot salt water. Fresh and cold salt water are also laid on. A large copper firemain extends from end to end of the ship, with frequent nozzles, and with a length of india rubber-lined hose at each nozzle. The cooking plant is extensive, there being four special ranges for native troops in addition to the galley ranges ordinarily arranged for passengers, European troops and crew. Large cold-storage rooms for meat and vegetables, cooled on the CO_2 gas principle, and an ice-making plant are also fitted. For ready use a provision store and ice room have also been placed on the main deck. A special cooled, filtered drinking-water service has been laid on to the pantries and bars. A soda-water machine has been fitted in a separate compartment on the orlop deck adapted for filling either syphons or glass bottles. There is also a bottle-washing machine. The storerooms, which are numerous and large, are constructed with stamped steel and netting to allow of a free circulation of air throughout them.

The hospitals for troops and women are fitted up complete, in accordance with the latest troopship practice, with all the usual cots, cot lifts, electric fans, portable electric lamps, electric kettles, electric heater for sterilizer, etc. A special hospital for the native crew has also been arranged and similarly fitted.

Admiralty practice has been adopted largely as regards ship's fittings and outfit, of which the following may be mentioned: Electric lighting, wiring and fittings, fire main, water-tight doors, sluice valves, night life buoys, compasses, voice pipes, hand pumps, masthead and gravity signalling lamps, semaphore, mess tables and stools, and helm signals. The electric plant consists of three compound closed-in engines, directly coupled to 50-kilowatt dynamos, also made by the Vickers Company. The voltage is, as in recent Admiralty practice, 100. The searchlight projector is portable, and is adapted for use in the ordinary way, or for being hung over the bows

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in a cage, for navigating in the Suez Canal. A powerful arc light for anchoring purposes in the Suez Canal is also supplied.

The ship has been fitted with Rothwell's patent davits and boat chocks, which greatly facilitate the getting out of the boats in a time of emergency. In accordance with the latest and best mercantile practice in this country and in Germany, the Stone-Lloyd system of controlling the watertight doors of the ship from the bridge has been adopted.

The ship has been designed for use as an auxiliary armed cruiser in time of war, and is adapted to mount eight 4.7-inch quick-firing guns and a number of guns of smaller caliber. Special watertight compartments have been arranged for the fitting up of magazines when desired, these spaces in ordinary trooping service being utilized as stores.

The *Dufferin*, unlike the previous troopers built for the Royal Indian Marine, has been completely fitted up in this country with all troop fittings, including mess tables and stools, sea-kit racks, accoutrement hooks, sleeping boards, hammock bars and hooks, mess racks, &c., so that on her arrival in India she will be ready, in all respects, for commissioning.—"Engineering," London, December 30, 1904.

A ONE-SIDE DRYDOCK.

A new floating dock has recently been added to the large number of similar docks in the port of Hamburg. At this port the floating dock is almost the only type of ship-repairing structure used. The particular dock illustrated differs from most others in Hamburg in some of its structural features. It was necessary to comply with some peculiar conditions. The river was so narrow at the place where the dock was to be used that no mooring rights could be obtained in the stream, so as to exclude any possibility of using the ordinary double-side type of dock, into which vessels enter by the end. The one-sided type was adopted accordingly.

The dock was designed to have a maximum lifting power

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of 11,000 tons, and is tied to the shore by nine sets of double parallel booms, of which two are A-shaped, so as to be able to lay hold of the dock also in its longitudinal direction, and to prevent any vibrations. To the same effect were provided two additional sets of booms, which are strongly cross-braced into pairs. The columns are built into concrete sockets formed by steel cylinders that are sunk into the river bed. These were filled with concrete after the soil had been removed. They are tied to the shore by means of steel ties fitted into concrete, all the columns being connected by means of lattice girders. The end and central booms serve as gangways leading to the top deck, while in addition to these, broader gangways lead down to the pontoon deck, passing through passages in the walls.

The over-all length of the dock proper is 508.4 feet. The bow is tapered, and has a length of 73 feet. The stern is approximately square on the land side, but the river side is tapered off to the same angle as the bow. The maximum beam of the dock is about 100 feet, the position of the keel blocks being such that the maximum beam of vessel to be taken is 76 feet, so that a working space of 4 feet between the side of the ship and face of the wall is allowed. The normal draught of the dock is 22 feet.

The machinery plant is located in the dock between the two longitudinal walls; it consists of eight sets of 18-inch centrifugal pumps driven each by a compound, horizontal engine, direct-connected with the pump shaft. According to the contract, the engines had to be of sufficient power to lift an 11,000-ton vessel in at most one and one-half hours. On the official trial they succeeded in lifting a 10,000-ton vessel in one hour, which may be called a very satisfactory result. The steam necessary for the operation of the engines is supplied from a set of boilers placed on shore, the steam being conducted to them through articulated pipes running along the shore booms. The whole pumping plant is operated from a central point, *i. e.*, the valve house, so as to be under excellent control. From this valve house is effected by hydraulic means

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the operation of the sluices, a small automatical semaphore indicating whether a particular gate is open or locked, while a water-level indicator accurately records in the valve house itself the exact amount of water in each compartment. The valve man is also in direct communication, by means of speaking tubes, with each engine, so as to control everything from his place.

In order to be able not only to lift vessels rapidly, but to berth them in as short a time as possible, special facilities had to be provided for. Powerful steam capstans were placed for this purpose on the top deck, whereas a series of mechanical side shores were provided in the wall of the dock, which can be screwed out to any desired extent, so that the centering of a ship over the keel blocks can be ascertained with safety.

In addition to the twenty ordinary mechanical side shores, ten very large and powerful hydraulic shores were provided, which are so arranged that, while advancing slowly, they can be withdrawn very rapidly, so as to enable ships entering the dock sideways to be placed in position rapidly and safely. The bilges of the ship are supported by means of mechanical bilge shores, consisting of watertight steel cases and worked from the top deck. They are fitted with a timber lining, one end being pivoted to the deck, while the other is raised by means of a steel rack and pinion actuated by a crank mechanism.

Both the engine and other rooms are provided with electric-light installation, arc lamps being carried on gallows on the top deck for outside illumination, while numerous flood boxes have been fitted, whence separate lamps may be taken for lighting any desired spot. For the washing of vessels, testing of tanks, and the thawing of the valves of the dock in winter, there have been provided cold and hot-water mains.—"Scientific American" Supplement.

HARMET STEEL PROCESS.

In recent years there have been recurring accidents in bursting of guns in firing. The guns were most carefully con-

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structed, from the best structural steel obtainable in the United States. This steel was tested most fully at every step of manufacture, and was only accepted when it met, in the physical tests, a very high standard of requirements. In endeavoring to determine the reasons for the occasional shortcomings, investigations were made in 1901, both as to what might be designated the intimate structure of the steel when in the ingot and after forging. The board on test of metals dissected two steel ingots, one being of carbon-steel and one nickel-steel. These ingots were found full of small interior cavities, resulting from the continual contractions of the metal when passing from the fluid to the solid state. There was also a considerable piping in the interior of the ingot.

Notable variations in chemical constituents in different parts of the ingots were found to exist. This was caused by what is called liquation and segregation, the occurrence of which appeared to be unavoidable in all methods in use. No amount of forging could eliminate the interior cavities referred to. They were simply "squeezed" together and lengthened by the act of forging. There being no adhesion in these flattened cavities, they are necessarily elements of weakness.

Investigations were made by the Ordnance Department, U. S. A., in 1901. Some rings, analogous to the rings forged for guns, were turned down. "Streaks" were found in many places, and it was obviously the result of the shrinkage cavities previously mentioned. Where these occurred the metal structure was necessarily weak. This was shown more fully when the rings were subjected to hydraulic pressure until they burst. The lines of breakage were where the streaks were most in evidence.

It is claimed for the Harmet process that it presents the economic advantage in that a larger proportion of the ingot is acceptable for gun, armor and other forgings admitting only of perfect metal. It is to be tested in this country, the Ordnance Department having ordered from abroad a 10-ton ingot for trial. The French government now accepts the Harmet process steel ingots with five per cent. of wastage, instead of

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twenty-eight demanded for the uncompressed steel ingots, and there is further economy, as it was found that the Harmet steel, with but two forgings, gave as good results in impact tests as the uncompressed steel gave in four forgings.

Summarizing, the claim for steel made by the Harmet process is: that it is more perfect in its physical structure, free from shrinkage cavities and piping; much more uniform in its chemical composition; gives twenty-five per cent. more available metal in the ingots produced, and requires a much smaller amount of forging to secure the best results. The process is used at Cammels' and Beardmore's in England. It is also now being installed by John Brown and Company and Firths, whilst other establishments are preparing to use it. It is in successful operation at the Oberbilk Works at Dusseldorf. It is, however, conspicuously absent in the steel establishments of the United States.—"Army and Navy Journal."

LONG RUN OF A TURBINE.

One of the most remarkable runs on record for a steam engine is that made by the 400-kw. Parsons turbine set exhibited by the Westinghouse Company at the St. Louis Exposition.

This turbine was started on the 20th of June and operated continuously until after the close of the exposition, on December 2d, the number of hours in operation being 3,962 at a speed of 3,600 revolutions per minute, making a total of 855,792,000 revolutions. An examination after shutting down showed the machine to be in excellent condition.

The set, in connection with a vertical and a horizontal gas engine, was used to supply light and power for the Westinghouse Company's extensive exhibit.

ARMOR CONTRACTS.

Contracts for the armor of the battleship *New Hampshire* and armored cruisers *North Carolina* and *Montana* have re-

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cently been awarded to the Bethlehem Steel Company and the Carnegie Steel Company; the former company will furnish that for the battleship and one cruiser, amounting to 5,666 tons, together with 94 tons of bolts and nuts, while the latter will supply 2,162 tons of armor for one of the cruisers.

The bid of the Midvale Steel Company was lower than either of the others, and this company would have been awarded the contract but for the fact that the plant is not as yet considered to be in a condition to furnish armor in quantities sufficiently large to prevent delay in the construction of the vessels.

TEST OF SUBMARINE BOATS.

A despatch from Newport News, Va., February 1, says: The Lake submarine boat, *Simon Lake X*, in her final test in James River today, broke the world's record submergence. She filled her superstructure in nine and one-half minutes. The record held by a French type is fifteen minutes. With decks awash she submerged in a depth of thirty-eight feet and arose to the surface in thirty seconds. The best time made by French type is one minute. The vessel's speed was as follows: Nine and one-half knots on surface; eight and one-half with decks awash, and six knots submerged. Commander Land, of the Argentine Navy; Captain de Chair, of the British Navy, and two other foreign officers were aboard during the tests.

The boat has been sold to a foreign government, and will be dismantled and shipped shortly. Inventor Lake left here today, and on February 7 will sail for Hamburg. It is stated that he has just secured contracts abroad amounting to more than $2,000,000.—"Army and Navy Register."

PORT OF NEW YORK, 1904.

During the year 1904 there arrived at the port of New York a total of 11,279 vessels exclusive of tugs, barges and small craft. Of this number 4,035 came from foreign ports,

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the greater number being of British registry, the second largest number of American registry, followed by German and Norwegian vessels. The remainder of the arrivals were from coastwise ports, 3,781 being from southern ports and 3,463 from eastern ports. Of the foreign arrivals 3,158 were steamers and 877 sailing vessels. Compared with the arrivals of the year 1903 there is a decrease of 975 vessels. The foreign arrivals increased twenty-two vessels, the southern arrivals decreased eighty-three vessels and the eastern arrivals decreased 914 vessels.—"Marine Review."

NEW DRYDOCK.

Contracts for a $150,000 floating drydock have been awarded to the Thompson Drydock Construction Co., of New York, by the New Orleans Shipbuilding & Drydock Co., the new dock being designed to hold vessels up to 5,000 tons and to replace the old Good Intent drydock. The new structure will be, after the navy dock, the largest in the Crescent City, measuring 330 feet long, 95 feet wide, and will draw at light draught 4 feet. It will be located just below Algiers Point, on the Algiers side of the river, where the company owns some 1,100 feet of frontage. The new dock is a direct consequence of the 2,500-ton dock recently completed at Mobile, which it was thought by New Orleans shipping men would probably draw considerable trade from them. About two hundred men will be employed when the dock goes into service.—"American Syren and Shipping."

WELDING TEST WITH THERMIT.

At the experimental test to decide the adaptability of the Goldschmidt thermit-welding process to marine repair work, held at the yard of the Dundam Towing and Wrecking Co., in Chicago last week, several hundred prominent marine and foundrymen saw a 7-inch anchor stock solidly reunited in less than half a minute. The process consists in fusing a chemical mixture of aluminum and iron oxide in which steel shav-

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ings have been mixed, in a mold fitting over the fracture. When burning the mixture attains a degree of 5,000 Fahrenheit. As an illustration of the fusing power of the mixture, a heavy steel plate was perforated by a jet of the blazing liquid as cleanly as though struck by an armor-piercing shell. The test was entirely satisfactory, and is believed to be of much use in marine lines.—"Marine Review."

(For a description of this process, see page 319, No. 1, Vol. XVI.)

OIL IN BRITISH NAVY.

Notwithstanding the conclusion reached by engineering experts of the United States Navy, after prolonged experiments, that the uncertainty of natural supplies of crude petroleum made it unadvisable to adopt that commodity for fuel purposes on warships, the British naval authorities have decided to install apparatus for the use of oil fuel on all the batteships of the Atlantic fleet, as well as on those of the new *King Edward VII* class. This policy has been agreed upon as the result of experiments carried out on two torpedo-boat destroyers with a new oil burner, the principle of which is kept secret, but which is said to generate forty per cent. more steam from each pound of oil than can be derived from a similar quantity of the best Welsh coal. The tests alluded to have demonstrated that oil fuel is no more harmful to a ship's boilers than coal of the best quality, while the labor required to handle it is less than half. The Admiralty are fully aware that oil can never be used as the principal fuel, but only as an auxiliary, owing to the uncertainty of supply, and because of this knowledge the installation of oil-burning apparatus on so many large vessels is significant of some concern regarding Great Britain's coal resources. A royal commission, which recently investigated the whole matter, has reported that England and Wales have still enough coal to last 450 years at the present annual output, the Welsh fields containing an estimated total of 3,397,-

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000,000, which is being mined at the rate of 18,000,000 tons a year, about six per cent. of which is used by the navy. The commission gives warning, however, that while the coal deposits are so rich as to afford no cause for apprehension, the cost of production is sure to increase with the increasing difficulty of mining, and that consequently it is a national duty to economize consumption by adopting all operative measures for the prevention of waste. It is evidently with that object in view that oil-burning apparatus is to be installed in a large number of important ships of the British fleet. If the new burner, for which so much is claimed, proves capable of producing steam for ordinary cruising purposes, economically, without involving increased wear and tear for boilers and machinery, the general subject of liquid fuel may become a matter of renewed interest in other navies.—"Army and Navy Journal."

TABLES FOR WORKING INDICATOR CARDS.

F. W. BARTLETT, LIEUT. COMMANDER, U. S. N.

These tables explain themselves and have been found of great use on board the *U. S. S. Chicago.* They may readily be changed to suit the number of cylinders or engines. The blank card with no name of ship or value of constants is intended as a suggestion for a Form to be added to those sent from the Bureau of Steam Engineering, made with perhaps six cylinders and three engines. They could then be used with all engines in the service. The same system of checks could be used. By adding across and up and down the checkage is made without the necessity of adding the columns twice for safety.

In case blanks are not supplied, it is possible to have Forms printed like the one marked for the *Chicago* on each ship. In case there is no printing press, the typewriter supplies its place, as many as eight or nine copies may readily be made.

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U. S. S. *CHICAGO.*

Indicator Card No. 190

Steam pressure.			
Cut-offs.			
	H.P.	I.P.	L.P.
S.			
P.			

STARBOARD ENGINE.

	H.P. out.	I.P. out.	L.P. out.	
P.	1.	1	1.	
N.	1.	1.	1.	
C.	1. 8.949	1. 9.305	1. 9.661	
I.H.P.	1.	1.	1.	Out.
	H.P. inb.	I.P. inb.	L.P. inb.	
P.	1.	1.	1.	
N.	1.	1.	1.	
C.	1. 8.930	1. 9.297	1. 9.657	
I.H.P.	1.	1.	1.	Inb.
Cyl.			S. Eng.	S. Eng.

PORT ENGINE.

	H.P. out.	I.P. out.	L.P. out.	
P.	1.	1.	1.	
N.	1.	1.	1.	
C.	1. 8.949	1. 9.305	1. 9.661	
I.H.P.	1.	1.	1.	Out.
	H.P. inb.	I.P. inb.	L.P. inb.	
P.	1.	1.	1.	
N.	1.	1.	1.	
C.	1. 8.930	1. 9.297	1. 9.657	
I.H.P.	1.	1.	1.	Inb.
Cyl.			P. Eng.	P. Eng.

BOTH ENGINES.

Check. Total I.H.P.

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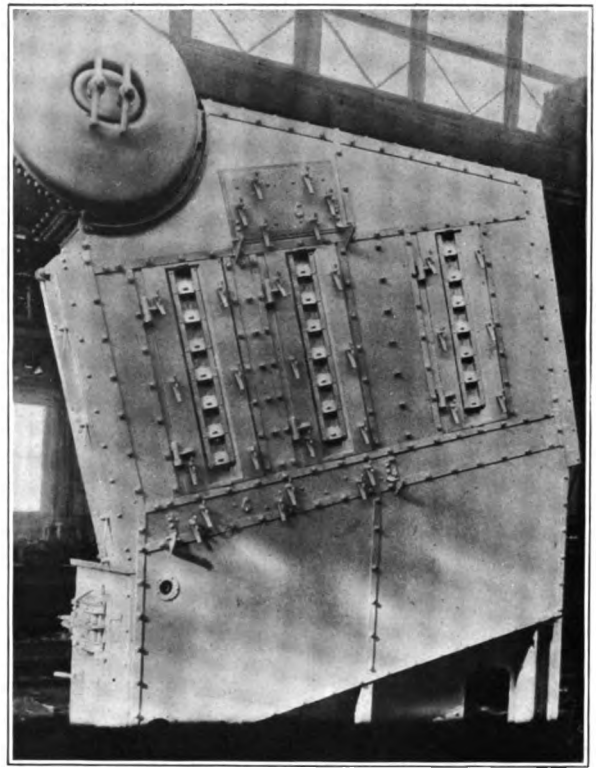

BABCOCK & WILCOX BOILER, BUILT FOR U. S. NAVAL ACADEMY.

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BABCOCK & WILCOX BOILERS.

The following notes, furnished by Commander W. N. Little, U. S. N., Inspector of Machinery for U. S. Navy at the works of the Babcock & Wilcox Co., Bayonne, N. J., will prove of interest, especially when considered in connection with the article on the latest type of this boiler by Lieutenant R. K. Crank, U. S. N., in No. 4, Vol. XVI, of the JOURNAL.

The notes and illustrations refer to a boiler recently built for the purpose of instruction at the U. S. Naval Academy.

The cast-iron baffle plates shown distinctly in the photograph, are held in place by friction between the tubes and plates, and are fastened to the side framing of the casing. Clamps between the side casing and the boiler tubes are shown in the picture explaining the supports to the casing. Usually the "mud" box connecting lower ends of all front headers is supported on a shelf bracket, not shown in picture, bolted to the vertical side box. The steam drum also acts as a girder suspending the front headers between the middle and side headers, the end and middle top nipples being surrounded by sleeves between header and drum upon which the drum rests. The hollow cast-iron baffle blocks resting on the lower 4-inch tubes are protected from the initial temperature of discharged gases of furnace by a course of fire bricks as shown, and in front of this baffle block may be seen the upper edge of the first baffle of fire brick resting on and filling space between front parts of the lower 4-inch tubes immediately over the furnace. To install the cast-iron baffles between tubes, the latter are separated very slightly by inserting and rotating a rod of flat iron between the tubes. In the right lower side casing, near front, is an opening over a brass plug screwed into the corner box abreast nipple between end of mud drum and corner box, to facilitate expanding said nipple.

As additional useful data in reference to these boilers so generally installed in our Navy, I offer the following table of water weights for one boiler of each vessel named.

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WEIGHTS IN POUNDS, COLD WATER.

Names of vessels.	Sections. No.	Length Ft.	Length Ins.	Lowest gauge cock.	Middle gauge cock.	Highest gauge cock.	Full.	Cold water temperature, deg. F.	Hot water at middle cock, pounds weight.*
Maryland	27	8	2	15,594	17,806	20,059	23,239	61.9	15,126
West Virginia	27	8	2	15,220	18,148	20,461	23,321	64.5	15,419
South Dakota	27	8	2	15,686	17,831	20,078	23,218	46	15,133
California	27	8	2	15,718	17,736	19,968	23,086	56	15,059
Tennessee	27	8	2	16,117	17,742	19,476	23,236	68.1	15,081
Washington	27	8	2	15,616	17,905	20,126	23,266	46 to 66	15,208
Nebraska	30	8	2	16,913	19,532	21,974	25,584	62	16,592
New Jersey	30	8	2	17,014	19,386	22,059	25,418	48	16,454
Rhode Island	30	8	2	16,891	19,225	21,725	25,263	56	16,324
Connecticut	25	9	2	15,309	17,353	19,503	22,503	74	14,760
Louisiana	25	9	2	15,449	17,691	19,683	22,463	68	15,105
Minnesota	25	9	2	15,167	17,384	19,548	22,488	57 to 67	14,759
Kansas	25	9	2	...	...	...	...	...	...
Vermont	25	9	2	...	...	...	...	...	...
Charleston	23	9	2	13,674	15,667	17,652	20,252	72	13,322
Milwaukee	23	9	2	13,663	15,606	17,614	20,274	60	13,326
St. Louis	23	9	2	13,769	15,663	17,723	20,343	74	13,384
Dubuque	15	8	2	7,522	8,818	10,037	12,033	74	7,535
Paducah	15	8	2	7,680	8,895	10,108	11,986	72	7,598
Indiana	20	7	11	11,140	12,825	14,510	16,870	72	11,155
Monterey	18	8	2	90,06	10,515	12,041	14,181	78	9,236

* Temperature of hot water, 411 degrees (steaming pressure 265 pounds), except in the cases of the *Indiana* and the *Monterey*, which temperature is 377 degrees (steaming pressure 175 pounds by gauge).

ADMIRALTY TRIALS OF A YARROW WATER-TUBE BOILER.

Through the courtesy of Messrs. Yarrow and Co., Limited, of Poplar, we are able to give below the results of trials made for the Admiralty on one of the Yarrow boilers for H.M.S. *Warrior*. Four-fifths of the boilers of this vessel are to be of the Yarrow water-tube type, the remainder being cylindrical boilers; but in the new boats water-tube boilers are to be used exclusively. The engines are to indicate 23,500 H.P., and there will be nineteen Yarrow boilers of four different sizes. The one tested was of the largest pattern, having a heating surface of 3,163 square feet, and a grate area of 53.35 square feet.

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SUMMARY OF EVAPORATIVE TESTS ON ONE OF NINETEEN YARROW BOILERS SUPPLIED FOR H. M. S. *WARRIOR*.

Heating surface, 3,168 square feet; grate surface as measured, 53.35 square feet; burning 24 pounds, 36 pounds, and 48 pounds of coal per square foot of grate; ratio, $\frac{H.S.}{G.A.} = 59.3$.

Carried out at Messrs. Yarrows' Works, Poplar, January 16, 17 and 19, 1905.

Coal per square foot of grate surface per hour.	Average steam pressure.	Average draught of air in the base of funnel.	Temperature of the feed water.	Pounds of coal burnt during the eight hours.	Pounds of water evaporated during the eight hours.	Pounds of water evaporated per pound of coal actually.	Pounds of water in separator.	Equivalent pounds of water evaporated from and at 212 deg. F. per pound of coal.	Water from and at 212 deg. F. per square foot of heating surface per hour.	Water from and at 212 deg. F. per square foot of grate surface per hour.
lbs.	*lbs.*	*in.*	*deg. F.*							
24	206.5	0.15	40	10,242	96,750	9.446	There was no water collected in the tank from the separator.	11.664	4.72	279
During the first four hours the back damper was closed, but the two center furnace doors were slightly open.										
36	208.8	0.775	38	15,365	136,310	8.871		10.974	6.65	394
48	208	1.03	38	20,487	182,800	8.922		11.038	8.93	530

Each trial lasted eight hours. Height of funnel, about 31 feet 6 inches to fire-bar level. When burning 24 pounds per square foot of grate surface, the steam jet in the funnel was not used. For the other two trials the draft was assisted by the steam jet in the funnel. The coal used throughout the trials was Ocean Merthyr. Readings, &c., were taken half-hourly, and the figures given are the mean results. On January 21 and 22 a leakage trial was carried out. The stop and safety-valve orifices were blank flanged, and a pressure of 210 pounds was maintained for 24 hours. At the end of this time no leakage was recorded.

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The specifications contained the following requirements as to coal economy:

Rate of coal consumption per square foot of grate per hour, pounds.	Equivalent evaporation from and at 212 degrees Fah., not less than pounds.
24	10
36	9½
48	9

These requirements, it will be seen from the tabulated results, were much surpassed at the trial. In the leakage test, which lasted 24 hours, blank flanges were placed over the stop and safety-valve orifices, and the pressure inside the boiler raised to 210 pounds per square inch. There was no leakage whatever.—"Engineering," London.

CONNECTICUT-LOUISIANA CONTEST.

The Department of Commerce and Labor has completed the investigation ordered by the House Labor Committee on the question of the desirability of the enactment of Gompers' eight-hour bill, and the report has been forwarded to Congress. An especially interesting feature of the report is the statistical data concerning the *Connecticut-Louisiana* contest, the *Louisiana* being built at the yard of the Newport News Ship Building and Dry Dock Co., Newport News, Va., and the *Connecticut* at the New York Navy Yard. The statistics on their face would seem to demonstrate the superiority of the eight-hour system when applied to ship building, but the Labor Bureau was careful to explain the reasons for this apparent advantage, which are so conclusive as to form a complete offset. The statement merely proves that the New York Navy Yard is working with the perfect system and straining every energy to prove that it can build a battleship. Every man and every minute is counting with it. The Newport News Co. is building the *Louisiana* precisely as it would build any other ship, without regard to the amount of work performed per man per hour. The committee does not attempt to make any record of cost, which is, of course, a vital consideration in any enterprize.

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Summarizing the results of hull construction, it is found that the average man at the New York Navy Yard accomplished as much every ten minutes as the average man at Newport News accomplished at every twelve minutes and twenty-five seconds, but in a day of ten hours the average man at Newport News worked in two-tenths of a pound more than the average man in the New Yord Navy Yard in a day of eight hours.

It was determined by the Newport News officials that the building of the hull offered the best and most available basis of comparison, and they contended and agreed to compile the weights of the structural material worked into the hull and the aggregate hours of all labor employed in the items of hull construction specified by them. The New York Navy Yard did the same for the *Connecticut*.

This comparison shows that the average production per man per hour on the *Connecticut* exceeded by 24.48 per cent. the average production per man per hour on the *Louisiana*, and in the opinion of the Labor Bureau explains why the progress on the *Connecticut* in the reports of percentage of work completed to the Bureau of Construction and Repair has kept pace with the percentage reports of work completed on the *Louisiana*.

The work on the *Louisiana* has been performed in the regular way and under normal conditions, while that of the *Connecticut* indicates the putting forth of unusual and extraordinary effort and energy. The reasons why this should be so called for further inquiry into prevailing conditions and elicited the following:

1. Higher rates of wages are paid at the Navy Yard than by private companies in Greater New York and elsewhere, and the rates of the latter average higher than companies elsewhere.

2. Employment the year around is steadier and more secure than in private yards.

3. The higher wages, shorter hours and steady employment attract the best grade of workmen to the Navy Yard, where a

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tacit recognition of an asserted economic theory prevails that the best workmen cannot be induced to work extra hard without larger pay than the average.

4. Prompt recognition of good work by advance in wages and promotion in grade.

5. A large waiting list of mechanics and others from private shops to select from.

6. The expectation or belief that if the *Connecticut* was built in record time the building of another battleship would be given the Brooklyn Yavy Yard.

7. A zeal generated by the general challenge of the country to the Navy Yard workmen to make good their claims in this test.

8. Prompt discharge for inefficiency.

9. Dismissal of workmen who could not or would not come up to a required standard of output in quantity and quality.

10. No restriction of output individually or collectively.

11. Loafing, soldiering or "marking time" not tolerated.

12. Workmen required to begin work the moment the whistle blows, and to continue working until the moment the whistle blows at quitting time.

13. Strict technical and exacting supervision of a high order of skill and experience.

14. A desire on the part of naval constructors and workmen to remove an impression of inefficiency growing out of former navy-yard construction of war vessels, before civil service regulations controlled employment there.

The report closes with the following pointed comment:

"Former investigations of industrial conditions, together with the facts brought out in this, lead to the following conclusions: Notwithstanding the remarkable achievement of the naval constructors and the workmen at the Navy Yard, probably unsurpassed in American industrial history, and the other notable examples in some of our great establishments in bringing out the best efforts and energy of their employés, it does not follow, when the facts presented by manufacturers

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are considered, that an eight-hour work day is possible under prevailing conditions, or that labor unions and shop management in general have reached a plane that would make it possible."—"Marine Review."

THE STEAM TURBINE FOR MARINE USE.

In his presidential address to the Institute of Marine Engineers, the Hon. C. A. Parsons, C. B., dealt with the past, present and future of the steam engine. He pointed out that James Watt, had clearly enunciated the principles governing the expansive action of steam, but it was impossible to apply these to the fullest extent in a reciprocating engine. In a warship at full power the ratio of expansion was only 5½ to 1, and in economical triple-expansion mercantile engines about 25 to 1. Nevertheless, with steam at 200 pounds pressure, and 27 inches vacuum, the volume at the lower pressure was 100 times that at the high, so that even in mercantile vessels the principle of expansion was not pushed to its limit. With a steam turbine the steam could be fully expanded, and trials with a number of different vessels showed that the turbines were economical, not only in the first little experimental *Turbinia*, running at 32 knots, but also in ocean-going passenger vessels of but 16 knots speed. From the experience gained he thought that ultimately the turbine would supersede reciprocating engines in all vessels of over 16 knots speed, and taking 5,000 indicated horsepower and upwards. It might also be used for vessels of but 13 knots speed, if of 20,000 tons and upwards, and perhaps even in slower vessels as time went on. As matters stood, about one-fifth of the total steam tonnage of the world was suitable for propulsion by steam-turbines; and as improvements were effected, this fraction would tend to increase. In vessels of the tramp class he suggested that a combination of reciprocating engines and exhaust steam turbines might prove suitable. In such vessels running at, say, a 10-knot speed, the number of revolutions must be low, because a certain disc area of propeller and a certain

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number of square feet of blade area were necessary in order to avoid excessive slip. It had been found by experiment that the pitch ratio could not be much less than 0.8 without incurring an excessive loss from the skin friction of the blades. On the other hand, to obtain a reasonable economy from the steam turbine, a certain bucket speed was required, and if the revolutions were low, both the diameter and the number of rings of blades must be increased in order to maintain economy. The result was that for a 10-knot speed the number of rows became inordinately great, and the weight and cost became excessive. By fitting reciprocating engines in conjunction with exhaust steam turbines the difficulty could be overcome, with an economy superior to the best tramp steam in existence. The turbine would receive the steam at about 7 pounds absolute, and expand it down to the condenser pressure. The gain effected would be 15 to 20 per cent., or of the same order as that obtained in the advance from the compound to the triple engines.—"Engineering," London.

SUBMARINE BOAT TAKES LONG PLUNGE.

One of the most unique tests ever held at Newport News occurred on January 16, when the Lake submarine boat *Simon Lake X* was given a submersion test in drydock No. 1, at the shipyard of the Newport News Shipbuilding and Dry Dock Co. The craft was floated into the drydock and the gate closed. Her ballast tanks were then filled with water and her hydroplanes set at the proper pitch, after which she was submerged. When near the bottom of the big basin a five-ton emergency weight attached to her keel was dropped from her hull, and with the water still in her ballast tanks the little craft rose to the surface of the water without the slightest hitch. The test was one of the most successful ever given the vessel, and the inventor, Captain Simon Lake, is highly pleased with the craft's behavior.

The emergency weight is carried by the vessel so that in case the other seven ways provided by Inventor Lake to

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make the vessel come to the surface of the water fail, the weight can be dropped and the vessel thus given a buoyancy of five tons, which is calculated to cause her to rise to the surface at once.

The *Simon Lake X* is almost completed, and the date for the government test will be fixed in the near future. Inventor Lake denies the reports circulated to the effect that *Simon Lake X* has been sold to Russia, and declares she will be used only in the government test.—"Nautical Gazette."

MARINE ENGINEERS' CONVENTION.

The thirtieth annual convention of the Marine Engineers' Beneficial Association was called to order by National President Frank A. Jones, at the Ebbitt House, Washington, D. C., on Monday morning, January 16, forty-five associations being represented by upwards of one hundred delegates. The sessions of the convention are not open to the public, but much work of interest to the membership was accomplished. It was decided to establish a national organ, and arrangements for its publication will be made by the national officers of the Advisory Board during the coming year.

The election of officers for the ensuing year resulted in the re-election of F. A. Jones, of San Francisco, as President, and of C. W. Robinson, of Washington, D. C., as First Vice President; E. L. Jenkins, of Cleveland, Ohio, Second Vice President; Geo. A. Grubb, of Chicago, Ill., Secretary; Albert L. Jones, of Detroit, Mich., Treasurer; and W. D. Blaicher, of Buffalo, N. Y.; W. F. Yates, of New York; G. F. Keating, of Boston, Mass., as members of the Advisory Board.

ANNUAL MEETING OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

The fiftieth meeting of the American Society of Mechanical Engineers, which was also the twenty-fourth annual meeting, was held in New York City on Tuesday to Friday, December 6

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to 9, 1904. A strong program of technical papers was considered, a program which dealt almost exclusively with machinery, tools and apparatus, or tests thereof. The discussion of Shop Management and similar general subjects, which has been such a large feature of the last half dozen meetings of the society, was absent.

The official business transacted at the meeting comprised three main items: the election of officers, the discussion of a proposed constitutional amendment, and the reception of two interesting mementoes, the gifts of members.

The ballot for officers resulted in the confirmation of the ticket presented by the nominating committee. The new officers are: President, Mr. John R. Freeman, of Providence, R. I.; Vice-Presidents, Mr. Samuel M. Vauclain, of Philadelphia, Pa., Mr. H. H. Westinghouse, of New York, N. Y., and Mr. Frederick W. Taylor, of Philadelphia, Pa.; Managers, Mr. George M. Brill, of Chicago, Ill., Mr. F. J. Miller, of New York, N. Y., and Mr. R. H. Rice, of Providence, R. I.; Treasurer, Mr. Wm. H. Wiley, of New York, N. Y.

At the opening session of the meeting, on Tuesday evening, December 6, President Ambrose Swasey delivered the presidential address, a brief but suggestive review of a few achievments in wonderful accuracy, under the title, "Some Refinements of Mechanical Science." Immediately following, Mr. C. W. Hunt addressed the society to present, on the behalf of an unnamed donor, a portrait of Professor John E. Sweet, of Syracuse, N. Y. Professor Sweet is one of the founders of the society, and to his activity and influence the success of the society's early years is largely attributed.

At the last session of the meeting Mr. James M. Dodge, of Philadelphia, Pa., presented to the society a bronze bust of Captain John Ericsson, the man who "twice revolutionized the science of naval architecture," as Mr. Hunt's presentation address had phrased it a few days before. The bust is a copy of a plaster bust made in the prime of the great engineer's life, a bust which was acknowledged by him to be an excellent likeness. The plaster bust had never been duplicated or re-

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produced in more enduring form until Mr. Dodge had this bronze replica prepared as a gift to the society.—"Engineering News."

EXPOSITION OF MILAN, 1906.

The following is published at the request of the Commission of the Exposition of Milan, 1906.

The Executive Committee, for the purpose of better attaining the object of uniting at the Milan Exhibition all the improvements which the human genius has made, as well in the artistic as in the industrial field, has brought to the notice of the Italian Minister of Agriculture Industry and Commerce, the necessity of formulating special legislative provisions (as has been done before in the case of all previous International Exhibitions) to give protection to everything that will be sent to the Exhibition from abroad.

The legal provisions invoked contemplate the derogation of the articles 50, 64 and 68 of the law on Italian patent rights, etc. These provisions should, for the objects exposed, suspend, first of all, the prohibition of introduction into the Kingdom of objects similar to those protected in the Kingdom, and limit the right to sequester infringements of articles bearing forbidden marks to the sole cases in which the person who provokes the sequestration holds the rights in the same country as the person who is to suffer the sequestration.

Further it should be sought to assimilate the showing of new inventions in the Exhibition to the promotion and utilization of inventions, and interrupt the expiration of patents, etc. Lastly, to the states who do not belong to the International Convention on Industrial Rights, there should be extended a protection to patentable objects exposed.

The Minister of Agriculture, Industry and Commerce has replied that this serious matter has already been taken into consideration.

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SHIPS.

UNITED STATES.

The following new vessels were put in commission during the year 1904:

Ohio—*Battleship.*—On October 4th. (For description of vessel and trial see page 1027, Vol. XVI.)

Denver—*Protected Cruiser.*—On May 17th. (For description of vessel and trials see pages 67 and 612, Vol. XVI.)

Des Moines—*Protected Cruiser.*—On March 5th. For description of vessel and trial see page 227, Vol. XVI.)

Tacoma—*Protected Cruiser.*—On January 30th. (For description of vessel and trial see page 1, Vol. XVI.)

Blakely—*Torpedo Boat.*—On December 27th. (See page 1076, Vol. XVI, for description of vessel and trial.)

Nicholson—*Torpedo Boat.*—On January 10th.

Tingey—*Torpedo Boat.*—On April 9th. (See page 50, Vol. XVI for description of vessel and trial.)

The following vessels were launched during the year 1904:

Virginia—*Battleship.*—By Newport News Shipbuilding and Dry Dock Co., Newport News, Va., on April 5th.

Nebraska—*Battleship.*—By Moran Bros. Co., Seattle, Wash., on October 7th.

Georgia—*Battleship.*—By Bath Iron Works, Bath, Me., on October 11th.

New Jersey—*Battleship.*—By the Fore River Ship and Engine Co., Quincy, Mass., on November 10th.

Rhode Island—*Battleship.*—By the Fore River Ship and Engine Co., Quincy, Mass., on May 17th.

Connecticut—*Battleship.*—At the Navy Yard, New York, on September 29th.

Louisiana—*Battleship.*—By the Newport News Ship Building and Dry Dock Co., Newport News, Va., on August 27th.

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California—*Armored Cruiser.*—By the Union Iron Works, San Francisco, Cal., on April 28th.

South Dakota—*Armored Cruiser.*—By the Union Iron Works, San Francisco, Cal., on July 22d.

Tennessee—*Armored Cruiser.*—By the William Cramp & Sons Ship and Engine Building Co., Philadelphia, Pa., on December 3d.

Milwaukee—*Protected Cruiser.*—By the Union Iron Works, San Francisco, Cal., on September 10th.

Charleston—*Protected Cruiser.*—By the Newport News Ship Building and Dry Dock Co., Newport News, Va., on January 23d.

Dubuque—*Gunboat.*—By the Gas Engine and Power Company and Charles L. Seabury & Co., Morris Heights, New York, N. Y., on August 15th.

Paducah—*Gunboat.*—By the Gas Engine and Power Company, and Charles L. Seabury Co., New York, N. Y., on October 11th.

New Hampshire, North Carolina and **Montana.**—The New York Shipbuilding Company has secured the contract for the building of the battleship *New Hampshire*, authorized by the Act of Congress approved April 27, 1904.

The contract for the two armored cruisers, *North Carolina* and *Montana*, provided for in this same Act, has been awarded to the Newport News Shipbuilding and Dry Dock Company.

A general description of these three vessels appears on page 1255, No. 4, Vol. XVI, of the JOURNAL.

Colorado—*Commissioned.*—The armored cruiser *Colorado* was placed in commission at the League Island Navy Yard on January 19, 1905. (For description of vessel and trials see page 1118, Vol. XVI, of the JOURNAL.)

Galveston—*Commissioned.*—The protected cruiser *Galveston* was commissioned at the Navy Yard, Norfolk, on February 15, 1905.

West Virginia—*Commissioned.*—The armored cruiser *West Virginia* was commissioned on February 24, 1905, at New-

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port News, Virginia. (For description of vessel and trials see page 59 of this issue.)

Submarine Boats.—The Electric Boat Company, of Bayonne, N. J., has recently secured from the Navy Department a contract to build two submarine torpedo boats of the Holland type.

These boats, which are to be of the latest design, will be constructed at the works of Fall River Ship and Engine Building Co., Quincy, Mass.

Newport—*Accident to.*—While *en route* to Sanchez, Santo Domingo, on January 30th, the U. S. S. *Newport* lost her propeller, due to breaking of the tail shaft. The accident occurred while about 400 miles southeast of Cape Henry, to which point the vessel proceeded under sail, arriving on February 9th, very heavy weather being experienced during entire trip.

The vessel has not yet been docked, and cause of accident is not at present known.

ENGLAND.

During the year 1904 the following vessels were commissioned for service:

Battleships.—*Cornwallis* on February 9th; *Queen*, April 7th; *Prince of Wales*, May 18th; *Swiftsure* and *Triumph*, June 21st.

Armored Cruisers.—*Euryalus*, January 5th; *Essex*, March 27th; *Lancaster*, April 5th; *Suffolk*, May 21st; *Cornwall*, and *Cumberland*, December 1st.

Protected Cruiser.—*Topaze*, December 6th.

In addition to the above the sloop *Cadmus* was commissioned on April 13th, and also eighteen torpedo-boat destroyers during the year.

The battleships *King Edward VII*, *Dominion* and *Commonwealth* were completed but not commissioned; other vessels completed were the protected cruisers *Amethyst*, *Diamond* and *Sapphire*.

Britannia—*Launch of.*—At Portsmouth, December 10, 1904, the battleship *Britannia*, of the *King Edward VII* class, was

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launched. Her principal dimensions are as follows: Length, between perpendiculars, 425 feet; beam, 78 feet; mean draught, 26 feet 9 inches; displacement, 16,400 tons. The following details are from "The Steamship."

The armament consists of the following guns: Four 12-inch breech-loading guns in barbettes forward and aft; four 9.2-inch breech-loading guns in barbettes on upper deck; ten 6-inch breech-loading 50-caliber guns of latest design in an armored battery on the main deck; twelve 12-pounder, 18-cwt., quick-firing guns on upper and shelter decks; two 12-pounder, 8-cwt., quick-firing guns on shelter deck for boat and field service; fourteen 3-pounder, quick-firing guns, three of which are also for boat service; two 0.303-inch Maxims, which are also intended for boat or field service. In addition, there are four submerged torpedo tubes, for which eighteen 18-inch torpedoes will be carried, besides six 14-inch torpedoes for the boats. An armored belt, varying in thickness from 9 to 2 inches and about 14 feet in width, extending to about 5½ feet below water line, will be worked for protection. The battery for the 6-inch guns will be protected by 7-inch armor, the guns being isolated by protective screens 2 inches in thickness. The 12 and 9.2-inch barbettes will be protected by armor varying in thickness from 12 to 4 inches. The ship will be propelled by twin-screws, each actuated by an independent set of vertical, triple-expansion engines, each with one high-pressure, one intermediate and two low-pressure cylinders of the collective power of 9,000 horsepower, giving an aggregate indicated power for both engines of 18,000 horsepower, the boilers being loaded to 210 pounds per square inch, the steam being reduced to a pressure of 205 pounds per square inch at the engines. With this power a speed of about 18.5 knots is anticipated. The boilers are of two types, these being three cylindrical and eighteen water-tube of the Babcock & Wilcox type. The former are arranged in one boiler room and the latter in three boiler rooms. The engines and boilers are being built by Messrs. Humphrys, Tennant & Co. The amount of coal car-

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ried at the normal draught is 950 tons, but stowage will be provided for about double that quantity.

Hindustan.—This first-class battleship has just completed her official trials. The *Hindustan*, the third of the modern British battleships built by Messrs. John Brown & Co., at Clydebank, is of the *King Edward VII* class, with a total length of 425 feet, and a beam and draught of 78 feet and 26 feet 9 inches, respectively. Her displacement is 16,350 tons, and her engines, also constructed by the builders of the vessel, are designed to indicate 18,000 horsepower, giving a speed of 18.5 knots. She is fitted with eighteen boilers of the Babcock & Wilcox type, and three of the cylindrical type.

Squirrel—*Launch of.*—Messrs. Workman, Clark & Co., Limited, Belfast, launched on December 21, 1904, the *Squirrel*, a coast-guard cruiser for service in the English Channel. The engines and boilers have been constructed by the builders at their Queen's road Works, and consist of a set of triple-expansion engines, supplied with steam from multitubular boilers working at a pressure of 155 pounds per square inch.

Argus—*Launch of.*—The 400-ton coast-guard cruiser *Argus* was launched by Bow, McLean & Co., Paisley, on December 6, 1904.

The *Argus* will join the fleet reserve; she was launched with steam up and proceeded immediately down the river for a series of trials.

Her battery consists of two 6-pounder, quick-firing guns.

Chelmer—*Launch of.*—The torpedo-boat destroyer *Chelmer* was launched by Thornycroft & Co. on December 8, 1904. She will have a speed of 25½ knots.

Wear—*Launch of.*—The new torpedo-boat destroyer *Wear* was launched on Saturday, the 21st ultimo, from the shipbuilding yard of Messrs. Palmers' Shipbuilding Company, Jarrow-on-Tyne. She is the second of three similar vessels which the firm is constructing for the Admiralty. The first of the three, the *Ure*, is nearing completion, and the third, the *Swale*, is on the stocks. They are of a stronger build than the earlier type. The *Wear* is 225 feet long and 23 feet 6

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inches broad. Her engines, fitted with Palmers' system of forced lubrication, will indicate 7,000 horsepower. She will carry two 12-pounder and five 6-pounder guns.—"Engineering," London.

Submarines—*Launch of.*—Two submarine torpedo boats of the "*B*" type were launched at the shipyard of Messrs. Vickers Sons & Maxim on January 23d.

Destroyers.—The new "River" class of low-speed torpedo-boat destroyers are to be followed by a type of destroyer able to outpace anything afloat. The Admiralty are credited with the wish to add to the Navy torpedo-boat destroyers able to steam from 33 to 34 knots, and, if this report be true, it is certain that the hostile criticism leveled against the 25-knot "River" class has been seriously considered at Whitehall. As has been pointed out before in these columns, it was not fair to condemn lightly-built swift boats merely because one or two of them, undoubtedly weak in structure, happened to be unable to withstand the fury of the sea. That it is not impossible to build fast destroyers with good sea-keeping powers Messrs. Yarrow or Thornycroft could easily prove. Safety must not be sacrificed to speed; but speed, it must be remembered, is at once the principal part of the defensive as well as offensive powers of this type of vessels. The projected 33-knot boats may be driven by reciprocating or turbine engines, and may burn oil or coal; but whatever combination is adopted the Admiralty are said to be determined to leave no stone unturned to get the fastest and safest boats of the class that British builders can turn out. If such destroyers were built, they must be "nursed" if they are to retain their speed, and must not be used as messengers in peace time on any and every occasion.—"The Marine Engineer," London.

Machinery for Cruisers.—The British Admiralty has placed orders for the machinery of three armored cruisers which are to be constructed in the Royal Dockyards. One of the ships will be engined by Scott's Shipbuilding and Engineering Co., of Greenock; another by Harland & Wolff, Belfast, and the third by Humphrys, Tennant & Co., London. These cruisers,

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to be known as the *Minotaur* class, are of 14,600 tons displacement, and in respect of guns and armor will be the most powerful cruisers in the British Navy, having four 9.2-inch guns and ten of 7.5-inch caliber. The speed will be 23 knots, to be attained by machinery of 27,000 indicated horsepower. There will be four cylinders in each engine, although the triple-compound system will be still adopted. The diameter of the cylinders is 40⅝ inches in the case of the high-pressure cylinder, 65½ inches in the intermediate and 74⅝ inches for each of the low-pressure cylinders, the stroke in all cases being 48 inches. The power is to be maintained with the engines making 125 revolutions. Babcock & Wilcox and Yarrow water-tube boilers will be adopted.—"Nautical Gazette."

New Scouts for the British Navy.—The performance of H. M. S. *Sentinel*, the first of the new scouts for the King's Navy, adds one more to the many successes in recent years of the Vickers Company; the vessel attained a speed considerably exceeding that of any vessel, either of naval or mercantile type, other than torpedo craft, and practically equal even to the rate reached at full power by the later torpedo-boat destroyers. The result indicates not only a very fine form of hull—of which a suggestion was given by the photographs of the vessel on the ways, which we reproduced on page 608 of Vol. LXXVII—but a high efficiency in each of the respective units of machinery—boilers, engines, and propellers. The actual speed realized was 25.24 knots, when the displacement of the vessel was 2,920 tons, and the power developed just over 17,500 indicated horsepower. Such a power in a vessel of this tonnage, and of a length of only 360 feet, has only become possible by the great advances in mechanical engineering.

Before referring in detail to the trials, some indication may be given as to the evolution and utility of the scout class. Effective reconnaissance work for any war fleet can best be done by high-speed well-armed cruisers, but the steady advance in size and fighting equipment of such vessels has so increased their cost that even a nation of great financial resources must consider whether it is not possible to have some of the subsi-

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diary duties, especially those involving numbers, performed by special and cheaper craft. The problem set is practically that met with in every-day commerce, and solved by the utilisation of telephones and messenger boys, so that the heads of the departments—the great thinking machines of establishments—may be able to concentrate their energy upon the most remunerative work.

The battleship represents the chief element of force in a war fleet; it has now been developed to a very high state of efficiency, which is only another way of saying that it costs an enormous sum of money. Britain is now paying over £1,400,000 for her ships-of-the-line; America, quite £1,500,000; and Japan proposes to lay down a ship the cost of which, with the armament—including four 12-inch, twelve 10-inch and twelve 4.7-inch guns—will involve still greater expenditure. The British ratepayer may as well realize now as later that there can be no finality in this matter, and when such a high authority on naval matters as Mr. Albert Vickers foreshadows the building of high-speed ships-of-the-line, with twelve 12-inch guns, it only becomes a matter of months when such experienced judgment will be carried into effect by the British Admiralty. We must therefore look forward to the building in the near future of battleships of the type indicated by Mr. Vickers, costing £2,000,000. The same advance in power and cost has taken place in respect of armored cruisers, and today, owing to the necessity of high speed in combination with effective broadside protection and high-power guns—and plenty of them—we are closely approaching a cost of £1,300,000 sterling. That being so, the question was tackled by responsible strategists as to whether relief could not be obtained by the building of special vessels for reconnaissance work, so as to conserve the energy of the cruisers until the enemy's fleet was discovered, when the armored cruisers might make a collective reconnaissance in force, piercing the enemy's screen effectually, and ascertaining the extent of the opposing fleet.

Such ordinary scouting work, being primarily intended to

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ascertain the position rather than the strength of the enemy, does not require the exercise of force to any great extent; and if four such scouts can be built for the price of one armored cruiser, as is now the case, the result is that the Admiral has four pairs of eyes, instead of one, to direct the action of the chief elements of force of his fleet. The Admiralty therefore decided to have a new type of cruiser, solely for scouting duty, and invited designs from the various ship-building firms throughout the country, laying down certain general conditions. Some firms, including the Vickers Company, suggested a hull of sound structural strength, so that a high speed might be realized under heavy weather conditions. This is an indispensable quality for reliable reconnaissance work. Others, however, submitted designs of lighter craft; but in the end all the vessels were brought up to the higher displacement. Eight have been ordered—two from each of four firms—Sir W. G. Armstrong, Whitworth & Co., Limited; Cammell, Laird & Co., Limited; the Fairfield Company, Limited; and Vickers Sons & Maxim, Limited; and the type has awakened very considerable interest.

It is easy to find fault with particular designs, because each student of tactics addresses the subject from the point of view with which he is either most familiar or most in sympathy. But it should never be forgotten that compromise is essential. In these scouts speed was the first consideration, and 25 knots is only a satisfactory *minimum* rate, as opposing armored cruisers may have a trial speed of 23 knots. We have indicated that it was not intended that the vessels should exercise force; the armament has been determined more with the idea of combating torpedo craft when used for scouting than for facing up to any large vessel. The scouts therefore have ten 12-pounder guns and eight 3-pounder guns, with two tubes on deck for launching 18-inch torpedoes. This is the armament set down by the Admiralty in their invitations for designs. Some would have had a more powerful armament, but any guns heavier than those fitted, yet too weak to combat a large cruiser of the enemy, would have been a waste of weight; it

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is is better far that the vessels should be able to run away for protection within the screen of their own squadron.

The question also has been raised as to whether these vessels have sufficient coal endurance. The details of size, load and speed must enter into this calculation. The addition of 100 tons to the bunker capacity of any ship involves a disproportionately large addition to the weight of the hull, which, again, necessitates increased engine power for a given speed. The scouts may seldom require to work at any great distance from the screen of the main fleet. High speed is required primarily for evading the enemy when discovered and also for informing the admiral of his location. Even this latter matter may not always involve the expenditure of coal necessary to attain high speed, because it would be idle for a scout to report the location of a squadron at any given time unless that time happened to be the moment when the admiral wished to engage it. Consequently the scout will require to hang on to the fleet while utilizing its wireless telegraphy to acquaint the admiral in command as to the situation. A scout may be used for dispatch duty, and the Admiralty have probably decided upon the radius of action with due regard to the fulfillment of such function, which demands a high speed. There is always the last resort of coaling from fleet colliers.

But the important point has been speed, as to the necessity for which all are agreed. It is not always realized by critics how much power high speed involves; every extra knot means an enormous addition to power. This was clearly brought out by the progressive speed trials which the Vickers Company carried out for their own information, and to add to that scientific data which enable them to predict with absolute precision results of a high order from any given set of conditions. It was found that while a speed of 19 knots was realized with a quarter of the full power, a corresponding addition to power only added then 3¾ knots; the third addition, of the same proportion of the total, increased the speed by 1½ knots, and the final fourth of the power added only about 1 knot. In other words, the increase from 24 to 25 knots cost as much in

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power as the first 19 knots. It is thus easy to understand that much could be done in the way of increasing armament, protection, or coal capacity, if 1 knot less speed were acceptable, since it would then be possible to reduce the weight of machinery by quite 20 per cent. Having determined that speed is the primary consideration, other elements in design must take a secondary place. In thus referring to the criticism of the type, we must not be assumed as adopting an apologetic tone, because, in view of the conditions which the designers were required to meet, the results in the case of the *Sentinel* are eminently satisfactory.

In the matter of protection the respective designers had a free hand; in two cases an arched deck of about 1½ inches in thickness is adopted; in two others the machinery is protected on the water line by broadside armor 2 inches thick. Here again the question arises as to excess of protection beyond the maximum required in view of the duty to be discharged. We do not say that the broadside armor is more efficient than the protected deck, because, with the curvature, the 1½-inch becomes more effective against lateral attack than with right-angle direct impact. Much can be said in favor of both systems. The real question is the efficacy of the protection afforded per unit of weight involved, and this is only a question which can be effectually determined in action. It must always be remembered that it will be the duty of the scout to run away. The protective deck certainly adds very materially to the strength of the ship, both longitudinally and transversely, so that it is an important element in the sea-worthiness of the ship. The question of the endurance of the machinery under adverse weather conditions in such light craft is an important one, and the more substantial the scantlings, and the more ample the bearings, the more likely is the ship to maintain her speed in heavy weather. The Vickers' design stands first in this respect. One of the firms has adopted torpedo-boat destroyer engines, two sets being fitted on each shaft, to run at 250 revolutions per minute, instead of 200 at full power, as in the *Sentinel*.

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The *Sentinel* is a vessel 360 feet between perpendiculars, with a beam of 40 feet, and, with all her sea stores and ammunition on board, her displacement is 2,920 tons. She is built with a high forecastle, but with a low freeboard amidships and aft, so that she will not be readily seen by the enemy. At the same time she has a very high bridge, so that she can watch the enemy's ship when hull down, so far as their vision is concerned. She has a foremast for signaling purposes, with a gaff for wireless telegraphy, as well as a truck semaphore. The machinery consists of two sets of four-cylinder triple-compound engines, well balanced. The cylinders are carried on cast-iron A columns at the back, and steel columns at the front, with a cast-steel bed plate. The propellers have three detachable blades, and, with the boss, are of bronze.

The contract trials differed from those specified for other ships of the Navy, and were much more severe. In the first place, the vessels of the class are required to steam for 96 hours at a cruising speed of between 10 and 12 knots; the coal consumption per knot on the latter half of this trial is to form the basis for determining the load to be carried in the bunkers on the full-power trial—when the vessel must take sufficient fuel to enable her to steam for 1,500 sea miles. The duration of this trial, and the acceptance of the last half, made it certain that, so far as the boilers—fire-grates, tubes, &c.—were concerned, the most adverse conditions as regards long-distance steaming would be realized. The *Sentinel* on this trial attained a high economy, and for the cruising speed it was found that each ton of coal was sufficient to propel the ship for 11 sea miles. This included the running of all the auxiliary machinery by night and day, and also the make up of all the feed water that was lost. For three-quarters of this trial the vessel experienced very severe weather, being driven for long periods in the teeth of a gale, so that the economy attained is the more remarkable.

The full-power trial took place on Friday, January 27, and followed upon the progressive-speed and other experimental

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runs to which we have referred; but it should be stated at once that these experimental runs were not involved by any degree of uncertainty as to elements of design, as on the first preliminary trial a speed of 25 knots was realized under the conditions which were evolved in the original design. In fact, throughout there has been a very remarkable assimilation between anticipated and realized results. On the full-power trial it was stipulated that the vessel should make six runs over the measured mile at Skelmorlie within a period of 1½ hours, and that on the subsequent 6½ hours the revolutions of the engines should correspond with those required on the measured mile for the guaranteed speed of 25 knots. The runs on the measured distance were made well within the period, notwithstanding the necessity of making wide circles at both ends of the mile, and of avoiding the traffic on the river. The six runs gave remarkably uniform results: the first run on the slack tide was made in 2 minutes 22.2 seconds; the second, also on the slack tide, in 2 minutes 22.8 seconds; the third, with the tide, in 2 minutes 21.2 seconds; the fourth, against the tide, in 2 minutes 24.4 seconds; the fifth, again with the tide, in 2 minutes 21.6 seconds; and the sixth, also in 2 minutes 24.4 seconds. It will thus be seen that the variation between the two runs on the slack tide was only three-fifths of a second; between the two runs with the tide, two-fifths of a second; the two runs against the tide being made in exactly the same time. The variations in the revolutions of the engines during this period were within practically one unit, and the mean number per minute was 206.2, while the power was just over 17,500. At the conclusion of this trial the vessel continued for 6½ hours at this power, and everything worked most satisfactorily. The Vickers boilers gave good results; they were, of course, worked under the closed-stokehold system, with an air pressure of 2½ to 2¾ inches of water. The water losses per 1,000 horsepower per 24 hours, due to all purposes, was about 3 tons, and the coal consumption was 2.1 pounds per horsepower per hour. These results show that not only from the point of view of speed, but of engine efficiency,

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the *Sentinel* marks a very high standard in naval construction.—"Engineering," London.

Submarine A-5—*Explosion on Board.*—In the harbor of Queenstown, Ireland, on February 16, an explosion occurred on board the submarine boat *A-5*, resulting in the death of four men and injury of fourteen others.

The cause of the explosion has not as yet been determined, but is thought to have been due to the gasoline on board.

Immediately after the explosion the vessel caught fire. A volunteer crew from the torpedo gunboat *Hazard* was sent on board, and almost immediately upon their arrival a second explosion occurred, blowing several overboard and injuring a number.

The injuries to the vessel are reported as slight. The *A-5* is a new vessel, having left the builders' yard on February 11th.

FRANCE.

The vessels completed for the French Navy during the year 1904, consisted of four armored cruisers, the *Gloire*, *Condé*, *Dupetit Thouars* and *Léon Gambetta*, also about eighteen submarines and five torpedo-boat destroyers.

Torpedo Boat Number 293.—The following data is from London "Engineering:" The boat is 130 feet long, 14 feet beam and 94.6 tons displacement, with 19.5 tons load—in fact, is as near like the reciprocating engine-propelled torpedo boats of the same type as was possible to build, modified only to suit the turbine machinery. Two Normand water-tube boilers supply steam at 250 pounds pressure. The hull was built by Augustus Normand & Co., and the engines by Parsons Marine Steam Turbine Company. A high-pressure turbine is fitted on the port wing shaft, and exhausts into the intermediate-pressure, driving the starboard wing shaft, which in turn exhausts into the low-pressure turbine on the center shaft. This arrangement is only for speed above 17 knots; for all lower speeds an additional high-pressure turbine is employed, and the ordinary high-pressure, which receives its exhaust, becomes the first

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intermediate. This cruising turbine, as it is called, is placed on the center shaft and coupled to the low-pressure turbine, and, when not required, steam is simply shut off from it and it revolves idly in a vacuum. The auxiliaries include two Normand condensers, Weir main and auxiliary feed pumps, air pumps driven by a reduction gearing from center turbine shaft, Normand feed heater and filter, etc. Reversing is accomplished by a set of reversing blades fitted at the aft end of the low-pressure turbine.

The eight-hour official consumption trial, at 14 knots, resulted in boilers taking 683 pounds of coal per hour, and the two-hours' continuous fuel speed runs gave an average of 26.205 knots and 26.663 maximum. An analysis of the trials compares favorably with the reciprocating-engined torpedo boats, for, with the cruising turbine, low coal consumption was attained such as not heretofore attained with turbines. In some torpedo boats the reciprocating engine has been fitted for cruising speeds with turbines for full power, which double arrangement is not altogether to be recommended.

Du Temple-Guyot Boilers.—The boilers of the two armored cruisers *Jules Ferry* and *Jules Michelet* will be of the *Du Temple-Guyot* type, with small tubes, but the majority of the other vessels under construction will be fitted with Niclausse or Belleville steam generators with large tubes.

Scouts.—The next program of construction is expected to contain particulars of a new class of protected cruiser, approximating to the scout type of the British Navy. They are to be of 3,000 tons displacement, with 26 knots speed, and the first batch will probably be laid down in the course of this year.

—"Page's Weekly."

GERMANY.

The following vessels were commissioned during the year 1904:

Battleships.—*Braunschweig* and *Elsass.*

Protected Cruisers.—*Hamberg*, *Bremen* and *Berlin.*

Six torpedo-boat destroyers have also been completed.

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München and Undine—*Launch of.*—The two protected cruisers, *München* and *Undine* were launched at Wilhemshaven on January 11th.

Notes.—"Ueberall" reports that the German battleship *Braunschweig* in her recent trials made an average of 18.43 knots during a five-hours' run. Her contract speed is 18 knots. In an endurance trial of eleven hours her engines developed an average of 11,588 horsepower. The average speed is not given, but with her engines developing 10,270 horsepower her speed was 16.41 knots, and 17.497 knots with 12,730 horsepower. The replacing of the rudder, which the German battleship *Elsass* lost on her trials, will take some time. The *Barbarossa*, of the *Kaiser* class, will be commissioned to take her place while she is in dock. The *Barbarossa* also lost her rudder in consequence of the breaking of her sternpost, and has been in dock under repair. Both vessels were built by the Schichau firm. At the trials of the German coast-defence ship *Aegir* after her reconstruction, the highest speed she attained was 15.27 knots. The results of the trials of the battleship *Schwaben* are still not published, but it is still hoped that she will be able to make 18 knots in deep water. The armored cruiser *Friedrich Karl* in a twenty-four hours' run averaged about 18.7 knots, her engines developing an average of 17,759 horsepower. The sea ran high and there was a strong wind against her. Later she made 20.5 knots in a six-hours' run, which still leaves something to be desired. The protected cruiser *Bremen* made 23.288 knots on the measured mile in deep water. In an endurance trial of ninety-three hours her engines averaged 124 revolutions. The speed is not given, but with 120.55 revolutions her speed was 20.327 knots, and with 135.88 revolutions, 22.466 knots. In a long run, therefore, she can be depended upon to maintain a speed of from 20.5 to 21 knots; but it is a pity she cannot carry more coal. The sea-going torpedo boat *S-123* made 28.3 knots on her trials. The same rate of speed was reached by *S-125*, which is the boat fitted with turbines, and built for a 27-knot speed. The armored cruiser *Prinz Adalbert*, while running

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under full steam with forced draft, burst the high-pressure cylinder of one of her engines. A new cylinder will have to be put in.—"Engineering," London.

Deutschland, *Launch of.*—The latest addition to the battleships of the German Navy is the *Deutschland*, launched from the covered slips of the Germania yard at Kiel, on November 19th. Her principal dimensions are the same as those of the *Braunschweig* class, of which the *Braunschweig*, only recently put into commission, and the *Hessen*, now almost completed, have been built by Messrs. Krupp, at the same yard. The length of these vessels between perpendiculars is 398 feet 6 inches; the maximum breadth, 72 feet 9 inches; and the draught, 25 feet 1 inch; their displacement being 13,000 tons. The armor belt is 9 inches thick amidships and 4 inches thick at the ends. The deck armor is 3 inches thick, and slopes at the sides to a junction with the lower edge of the belt. The side armor of the citadel and battery-deck casemate is, respectively, 5½ inches and 6 inches thick, the casemate being covered in by a protective deck. There are two conning towers, the one at the fore being protected by 12-inch armor and the aft one by 5½-inch armor.

The main armament consists of four 11-inch guns mounted in a pair of turrets fore and aft, and protected by 11-inch armor. In the central casemate are ten 6.7-inch guns, and four more guns of the same caliber are contained in separate casemates. There are twenty-two 3.5-inch quick-firers mounted on top of the casemates. The fighting tops contain four automatic machine guns of 1.5-inch bore, for protection against the attacks of torpedo boats, and four other machine guns of 0.3 inch. Six 18-inch submerged torpedo tubes are fitted—four broadside, one at the bow, and one at the stern of the vessel.

The *Deutschland* is propelled by three sets of triple-expansion engines, aggregating 16,000 indicated horsepower, and expected to give a speed of 18 knots. Steam is supplied by six cylindrical boilers and eight water-tube boilers, the latter of the Schulz-Thornycroft type. The normal coal-carrying

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capacity is 700 tons, but 900 tons more can be stored in the reserve bunkers. The double-bottom compartments are fitted for the carriage of 200 tons of tar oil, which is also intended as fuel.

The vessel will carry the Admiral's flag, and will be manned by 35 officers, 35 warrant officers, 16 ensigns and a crew of 650 men.—"Engineering," London.

ITALY.

During 1904 the battleship *Benedetto Brin* and two destroyers were commissioned for service.

Delfino.—The Italian submarine boat *Delfino* is built of steel plates 1.2 inches in thickness. She is cigar-shaped, her length being 78.4 feet and her beam 9.5 feet. Her displacement varies, according to the extent of her submersion, from 95 to 107 tons. Her engines are worked solely by electricity furnished by 300 accumulators. She has three propellers—one aft for movement ahead or astern, and the other two above for the work of submersion and emersion. The little turret is glazed so that a lookout may be maintained when the boat is submerged. Her sole armament consists of two torpedo tubes forward. Her oxygen supply is not sufficient for officers and crew more than twelve in number.—"Scientific American."

MEXICO.

Bravo and Moreoa.—There were recently delivered to the Mexican government two twin-screw gunboats, the *Bravo* and *Moreoa*, built at the Oderos shipyard at Sestri Ponente, near Genoa. The general dimensions are as follows: Length, 250 feet; beam, 34 feet; depth, 14 feet; displacement, 1,280 tons; designed speed, 17 knots. The vessels are very light draught, for navigating the shallow rivers of the Mexican coast, while, at the same time, the bunker capacity is sufficient to carry the vessel 5,000 miles at a speed of 10 knots. The propelling machinery consists of two sets of three-cylinder, direct-acting, triple-expansion engines of 2,600 I.H.P. Steam is supplied by two water-tube boilers.

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There are two 4-inch R.F. guns of Bethlehem make, which fire directly ahead and astern respectively, and have a total arc of fire of 150 degrees; four 2¼-inch R.F.G., Schacidercant type, are fitted on broadside and one on the bridge. The 4-inch guns are supplied with ammunition by electrical hoists capable of delivering 24 charges per minute. Electrical hoists are provided for the other guns also. The crew will be composed of 26 officers and 90 men, and in addition quarters are provided on board for about 270 troops. There are six boats, two of which are steam launches, for quickly disembarking the troops. Fresh air is supplied to the quarters by electric fans, and a large refrigerating plant is installed. The *Bravo* will proceed direct to Mexico as soon as she has completed her trial trip.—"Marine Engineering."

THE NETHERLANDS.

The Dutch Navy budget for 1905 provides for the completion of the battleship *Tromp*, the four torpedo boats, *G-1–4* (of which *G-1* was launched on December 17th and *G-2* on December 28th, by the Schelde Company), and one submarine. A new battleship, "*E*," is to be laid down and four more torpedo boats, sisters to *G-1*, are to be begun. The torpedo boats *Tangka*, *Wajang* and *Semeroe* have made 25.6, 24.7 and 24.6 knots respectively on trial.—"Page's Weekly."

RUSSIA.

During the year 1904 the following vessels were completed and put into service:

Battleships.—*Kniaz Suvaroff*, *Orel* and *Borodino*.

Protected Cruisers.—*Izumrud*, *Jemtchug* and *Oleg*.

Naval Program.—"The Emperor has sanctioned the expenditure of £160,000,000 for rebuilding the Russian Navy." In these bald words the St. Petersburg correspondents announced the new scheme for the replacing of the squadrons which Russia has lost in the war. The program is to cover ten years, and many of the vessels will be built abroad. The building program is to consist of eight battleships of the

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Kniaz Suvaroff type, eight of the *Andrei Pervosvannui* class, six armored cruisers, twelve protected cruisers, fifty destroyers, and one hundred torpedo boats.

The Third Pacific Squadron, which is fitting out in great haste at Libau, will take practically every remnant of the Russian fleet from European waters, with the exception of the Black Sea Fleet, which is penned up behind the Dardanelles. Whether the squadron under Admiral Mebagatoff will be ready to start in another fortnight is doubtful, but urgent commands have been issued for it to do so.—"Page's Weekly."

SWEDEN.

Naval Program.—The Naval Program of 1901, which was to have been completed by next year, has fallen so much in arrears that an increased grant has been asked for for 1905, in order that the program may be completed in 1907. The following is the proposed allotment of this grant. The sums asked for are as follows: For completing the ironclad *Oskar II*, £140,100; first instalment new ironclad of *Fylgia* type, £123.000 (total cost, £385,700); completion of torpedo-boat destroyer, £33,400; two torpedo-boat destroyers, £133,500; first instalment five large torpedo boats, £55,100; nine small torpedo boats, £99,100; training vessel, £11,600; reconstruction of *Loke*, £29,800; ditto of *Hildur*, *Bjorn* and *Gerda*, £28,050; three guard boats, £14,100.—"Page's Weekly."

MISCELLANEOUS.

Dredge St. Johns.—The sea-going suction dredge *St. Johns*, one of two vessels being built for the U. S. Engineer Corps, has recently been given a successful trial at sea.

These vessels are 200 feet long, 40 feet beam, with a depth of 23 feet 2 inches; they are self-propelling, with wooden hulls. The engines are of the vertical, inverted, compound type and develop about 1,200 indicated horsepower. The boilers, two in number, are the ordinary Scotch type. The pumping machinery on each vessel consists of two centrifugal pumps driven by a vertical, inverted, compound engine of about

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700 horsepower. The vessels have in addition the usual auxiliary machinery. The suction pipes are 18 inches diameter (swiveling amidship), and are constructed so that they can be lowered or inclined at the free ends where heavy scraper nozzles are attached. These dredge the bottom of the harbor, and sand is pumped up by means of powerful centrifugal sand pumps, which discharge the heavier material into binns constructed in the hold of the vessels. These binns have a capacity of 1,600 tons, and can be filled in about forty-five minutes. When under way the suction pipes are drawn up and the vessel proceeds to sea, where the sand is discharged through numerous sluices in the bottom of the hull.

Hydraulic Suction Dredge.—The Polson Iron Works, Toronto, Ont., is building a hydraulic suction dredge for the Dominion government, which is intended for service on the St. Lawrence channel between Montreal and Quebec. The dredge is somewhat similar to the *J. Israel Tarte*, built by the Polson Iron Works in 1902. The principal dimensions are: Length over all, 127 feet; breadth, 32 feet; depth at side, 9 feet; draught 5 feet. The hull will be of scow model, with square bow and rake stern, for the purpose of towing easily. It is to be constructed of mild steel, divided into bulkheads, and fitted with bunkers to have a capacity of 100 tons of coal. The main deck will be of 3 by 4-inch British Columbia pine, the upper deck of 2-inch British Columbia pine, and there will also be a light shade deck over the hoisting engine. A cabin for officers and crew will be provided; it will be steam heated. Two boats are also to be supplied with the dredge.

The machinery will consist of a direct-acting, inverted, triple-compound, three-crank, surface-condensing engine, having cylinders 14, 22 and 36 inches in diameter by 21-inch stroke, and built for a working pressure of 160 pounds a square inch, and is calculated to develop 600 I.H.P. when running at 190 revolutions a minute. The main dredging pump will be a centrifugal one, with the suction and discharge pipes 24 inches in diameter; and the feed and bilge pumps will consist of two independent horizontal, duplex,

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outside-packed plunger pumps 7¼ by 4½ by 10 inches. The hoisting engine for raising and lowering the suction pipe and swinging the boom will be located on the main deck forward, and will be a 10 by 12-inch, two-cylinder, double-acting, independent-type engine, with link motion to allow operation in either direction. The steel boom for raising and lowering the suction pipe will be located at the suction end of the dredge, and will be about 65 feet in length. The suction pipe will have an internal diameter of 24 inches, and will be fitted with a cutter head at the outer end. To excavate the material there will be a rotary steel cutter. The steel head of this will be of suitable shape, and will have attached to it a renewable steel block of such a shape that it will convey the material taken from the bed of the river into the pipe. The outside diameter of the head is to be large enough to enclose the whole of the mouth of the pipe. This head will be driven by a steel shaft carriad along the whole length of the pipe, and receiving its power from a separate engine, by suitable steel gearing. The engine to supply this power will be a two-cylinder, double-acting horizontal engine, 12 by 12-inch stroke, arranged to run in one direction only. The discharge pipe will also be 24 inches in diameter, and will be arranged for discharging the material into scows. The capstans will be operated by two 6 by 8-inch double-acting engines, which are to be located on the main deck forward of the hoisting engine. Steam will be supplied to the various engines by two marine return-tubular boilers, allowed a working pressure of 160 pounds to the square inch. They will be 12 by 12 feet 6 inches, and will have three furnaces, each 35 inches in diameter, and a grate surface in each boiler of 48 square feet. There will be two smoke stacks, each 42 feet in height from center of boiler.—"Marine Review."

Murikiku.—Messrs. William Simons and Co., Ltd., Renfrew, have just launched, complete with steam up and ready for work, a very powerful dredging vessel of their latest design. The boat is named the *Murikiku*, and she has been built to the order of the Agent General for New Zealand. Besides being

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fitted with a special set of buckets for dredging rocky ground, the vessel is fitted with a sand-suction pump. The buckets as well as the suction pipe are made to dredge to a depth of forty feet. The hopper is arranged in such a manner that the dredgings can be relifted from the hopper and discharged over the side by a special patent arrangement of the builders, and delivered ashore through a long line of floating pipe for land reclamation, or into barges moored at the side of the dredger. Ordinary hinged doors are also fitted to the vessel's own hopper, so that the material can be deposited at sea when required. The machinery consists of two sets of triple-expansion engines and two cylindrical boilers of 160 pounds working pressure.—"Page's Weekly."

Ice-Breaking Steamer.—The second of the ice-breaking steamers ordered by the Dominion government for the St. Lawrence, has reached Quebec. She was built in Scotland, the principal dimensions being: Length, 245 feet; breadth, 40.5 feet; depth, 18 feet, with a gross tonnage of about 1,350 tons. The vessel is specially strengthened in order to resist pressure, the shell plating at bows and in other parts being of double thickness, with intermediate frames extending fore and aft. The keel, stem, stern post, propeller brackets and rudder are extra strong and heavy, and made of nickel-steel. On the stern post a cast-steel knife is fitted to break ice when working astern, and to protect the rudder. She is propelled by twin screws, and is expected to develop a high rate of speed.—"Nautical Gazette."

Turbine Steamer.—The fleet of nine steamers which the Belgian government maintains for the cross-Channel service is to be increased by the addition of a turbine steamer with a speed of 23 knots. It is said that Belgium intends to replace gradually all the old steamers of a speed of 19 knots by turbine steamers similar to that to be introduced shortly.

Fire and Salvage Float.—A new type of combined fire and salvage float has been constructed for use on the Manchester Ship Canal. The craft is of great power for both the functions for which it has been designed. For fire-extinguishing

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work the vessel carries three large monitors, each capable of throwing a solid 2½-inch jet of water to a maximum height of 250 feet at full pressure, and there are also twelve outlets for hose connections, each with main-gauge instantaneous couplings. The monitors are placed forward, amidships and aft, respectively, and each is fitted with wheel and worm gear so that the jet can be directed at any angle. A total volume of 3,000 gallons of water can be discharged per minute. For salvage purposes the pumps have a discharging capacity of over eighteen tons of water per minute. The speed of the craft is eight knots per hour. The float has been designed more especially for coping with conflagrations among the great warehouses on the banks of the canal, wherein is stored highly inflammable freight.—"Scientific American."

Police Boat.—W. A. Boole & Son, San Francisco, are building a 125-ton gasoline schooner for the German government. This vessel is for service in the South seas, where it will be used as a police boat in the ports of the Kaiser's tropical possessions. The schooner will be 106 feet long, and, although equipped with a 125-horsepower gasoline engine, will be heavily sparred and have wings large enough to insure independence of auxiliary power. Commodious officers' quarters will be provided in the after end of the vessel, and the space usually set apart for cargo will be fitted up for the use of native police. The vessel has a very symmetrical model and all her lines are very handsome and graceful. She will have a speed of fifteen knots per hour. —"Marine Review."

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MERCHANT SHIPS.

THE NEW CUNARD LINERS.

The Cunard Steamship Line has been regarded, since its inception sixty-five years ago, as a great national institution, largely contributing, as it has done, not only to the development of British shipping supremacy, but also to the evolution on sound lines of marine engineering. The agreement entered into between the Government and the Company about a year ago was therefore regarded with great satisfaction, as it insured, irrespective of cost, the undoubted advantage which accrues from the maintenance of maritime supremacy especially in respect of high speed, and will place at the disposal of the Admiralty ships of great utility for service in war times. Under the agreement the Cunard Company are, as is well known, building two steamers, 785 feet long, to attain a speed of 25 knots. But Lord Inverclide, and his colleagues on the directorate, recognized from their long experience that, while there is a large number of trans-Atlantic passengers who will, at all costs, choose the fastest ship, there are others, probably a great majority of the voyagers, who are satisfied with a moderate speed, provided there is certainty of at least the same measure of comfort as is afforded in the high-speed liners. When fixing the general principles of the high-speed mail ships, the Cunard Company decided upon a great step in advance in connection with what has hitherto been regarded as the "intermediate" class of ship, and orders for two of these vessels, the *Caronia* and *Carmania*, were placed with Messrs. John Brown and Company, Limited, of Clydebank. It is easy to demonstrate that these two ships are of the first class in every element of design, excepting only speed, while the fact that even in this respect they are excelled by only eight or nine ships on the Atlantic gives them a high place among the world's ships.

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There are only three or four vessels of greater tonnage than the *Caronia* and *Carmania*, and the difference is due to the aim to carry more cargo in the larger vessels. The *Baltic*, which is the biggest ship afloat, is 48 feet longer, and has a gross tonnage about 2,800 tons greater, while in view of her fuller lines and greater cargo, she will in service have a displacement considerably higher, but her speed on the Atlantic is 17 knots. The over-all length of the *Caronia* and *Carmania* is 678 feet, and the molded breadth 72 feet, making the displacement 29,800 tons when the vessel is fully loaded to a draught of 32 feet. In order to attain a speed of 19 knots under economical conditions a smaller coefficient of fineness was decided upon than has been adopted in previous so-called intermediate steamers, it being under 0.7, which compares with about 0.55 for some cruisers; but with the great length possible in ships for merchant service, the naval architect does not require to fine the ends of his ships to the same extent as in the short, beamy armored vessel. Thus it may be assumed that, even in the 25-knot ship, the coefficient will be somewhere about 0.6. But, to return to the question of size, the molded depth of the *Caronia* is 52 feet, the height from keel to navigating bridge is 90 feet and to the top of the funnels 144 feet. In her construction 12,000 tons of steel were required, with 1,800,000 rivets. Many of the plates were 1⅛ inches thick, 32 feet long and 5½ feet in width, quadruple riveted with 1⅛-inch rivets by hydraulic pressure. She may carry 3,450 tons of water ballast, 12,000 tons of cargo, and 300 first, 350 second, 1,000 third and 1,000 steerage passengers. These, with the crew of 450 officers and men, make an aggregate of 3,100. As to the superior character of the accommodation provided for all classes, and the thousand and one details to insure comfort, we hope to deal later, when we publish further illustrations. For the present we may only say that there is nothing wanting in any department of the ship.

Chief interest, however, in connection with the *Caronia* and *Carmania* centers in the propelling machinery. The

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former is fitted with quadruple-expansion engines, driving twin screws; for the latter, to be launched on the 21st inst., there is now being constructed at Clydebank an installation of Parsons steam turbines to drive three screws. Opportunity will thus be given of testing the comparative economy of the systems in the Atlantic service; and while doubt as to the potentialities of the newer prime mover have almost disappeared except in the minds of most conservative engineers, the results will always be valuable, so that Lord Inverclyde and his company are to be felicitated not only on the bold step of adopting the system for the 52-knot ships, but particularly for affording means of a thoroughly reliable test in the most severe of steamship services. It may be claimed that, so far as the reciprocating machinery is concerned, full justice will be done as an outcome of the experience of the Clydebank staff, while at the same time a year's continuous research in a special turbine experimental department, and in the ship-model and propeller-testing tank at Clydebank, has enabled Messrs J. Brown and Co., to determine many uncertainties in connection with details as to the design, and especially as to the manufacture, of the turbines. The time has barely come for reviewing this experimental work, but one or two points may be dealt with.

In the first place, however, reference must be made to the reciprocating engines. Standing in the engine room, one could not help being impressed, not only with the great size and suggestion of strength, but also with the easy motion of these engines. There is no longitudinal bulkhead between the two engines, and their appearance when running at ninety revolutions was therefore the more impressive. The total height from the center of the shafts is 30 feet, and from the base of the bed plate 36 feet. The diameters of the respective cylinders are 39 inches, 54½ inches, 77 inches and 110 inches. The latter, with one exception, is the largest cylinder yet adopted for vertical engines. The stroke is 5 feet 6 inches, and the length of the connecting rods is 12 feet between centers. This length is conducive to efficient working, and in

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this respect the merchant ship has a great advantage over the warship. In the latter, with engines of corresponding power, the stroke would be only 4 feet 3 inches, with a connecting rod 8 feet 6 inches between centers. Again, the weight allowed for machinery is probably double that placed at the disposal of the warship builder; naval machinery requires to make 40 per cent. more revolutions for full power.

While the naval engine is only expected to run at full speed in cases of emergency, and even then only for short periods, the *Caronia's* machinery has been built to run year in and year out at full speed, irrespective of weather, and consequently many conditions had to be taken into account in the proportioning of parts which do not apply in the case of naval machinery. Merchant-ship boilers are thus made of sufficient capacity to maintain the full power continuously, even when a certain number are shut down for cleaning fires. In the naval service the boilers are usually proportioned to maintain continuously four-fifths of the power, the remainder required for full speed being got under pressure. The *Caronia* boilers, which, along with the engines, were fully described on the occasion of the launch,* have thus 2.35 square feet of heating surface per unit of power, Howden's system of forced draft being used, as against 2.7 square feet for natural draft in cruisers; the power per unit of grate area is 17.3 indicated horsepower and 13 indicated horsepower respectively. The *Caronia's* machinery has been evolved from that of the *Saxonia*, which was tested by the Naval Boiler Committee, and required only 13.4 pounds of steam per horsepower per hour, against 16 pounds in the naval ships tested; while the boilers, with Howden's system of forced draft, generated 12.33 pounds of steam per pound of coal, as compared to 9½ pounds to 11 pounds in the naval ship. The coal consumption per unit of power was 1.29 pounds. There is every promise that in the *Caronia* the results will be at least as good, so that the standard set for comparison with the turbine installation in the

* See "Engineering," Vol. LXXVIII, page 91.

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Carmania will do full justice to the reciprocating type of machinery.

As regards the *Carmania*, the only difference will be in the propelling machinery. The form of hull, except so far as modified for the propellers, and the boiler installation, will be exactly the same. There will be, as in all recent installations, three shafts, the center shaft being driven by the high-pressure turbine, while the side shafts will be worked by the low-pressure system, with the go-astern turbine at the after end fitted inside the low-pressure casing. The space occupied, so far as area is concerned, will be the same as in the *Caronia;* there will be a saving in weight by the adoption of the turbine system of about 5 per cent. The boiler pressure will be reduced from 210 pounds, as in the *Caronia*, to 195 pounds in the *Carmania*. The turbines will take steam at an initial pressure of 165 pounds, as against the 200 pounds in the quadruple-expansion engines. Owing to the high vacuum desirable in the turbine system to ensure the fullest economy, the condensing surface, which in the *Caronia* is 27,030 square feet, will be increased in the *Carmania* by 20 per cent., while the capacity of the centrifugal pumps will be increased by about 50 per cent., so that the amount of circulating water that they will be capable of discharging will be from 50 to 60 times the weight of feed water, as compared with a ratio of from 25 to 30 times in the *Caronia* installation.

The trials of the *Caronia*, which took place on Saturday, the 4th instant, and Monday, the 6th instant, gave very satisfactory results. In the first place a series of progressive speed runs was made over the measured mile at Skelmorlie, on the Clyde, with the following mean results:

Revolutions.	Power. Indicated horsepower.	Speed. Knots.
69.6	10,400	15.72
74.5	12,350	16.61
80.9	15,500	17.76
85.2	18,900	18.8
89.	21,600	19.51

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After these progressive runs the ship continued at full power for some time, and the highest results during the performance on Saturday, so far as engines were concerned, were as follow: Revolutions of port engine, 90.9; power, 11,550 indicated horsepower; revolutions of starboard engine, 91.2; power, 11,527 indicated horsepower. The collective power was thus 23,077 indicated horsepower. The official full-power trials commenced at noon on Monday. The mean result of four runs on the measured mile was as follow: 89.2 revolutions; 21,870 indicated horsepower; 19.62 knots; and this speed and revolutions were maintained for a total continuous period of 13½ hours. Indicator cards were taken at regular intervals during the trial, and the highest power recorded was about 23,500, the results of the previous trial on the 4th instant being thus verified in all essential details.—Extract from article in "London Engineering."

Virginian, *Launch of.*—On December 22, Messrs. Alexander, Stephen & Sons, Limited, Glasgow, launched the new turbine steamer *Virginian*, 11,200 tons, which has been built for the Allan Line Steamship Company, Limited, of Glasgow. This vessel, with the *Victorian*, now completing at Belfast, will be the first turbine liners to cross the Atlantic. The *Virginian's* dimensions are: Length, 540 feet over all; beam, 60 feet; and depth, 41 feet. She has accommodation for about 1,650 passengers—470 first class, 240 second class and 940 third class. The vessel is specially designed for the conveyance of the Canadian mails and passengers; no less than five decks are fitted up for the accommodation of the latter. She has also considerable cargo space in the holds, fore and aft, one of them being fitted up and refrigerated for the reception of perishable cargo, chilled beef, etc. The first-class dining saloon at the fore end of the bridge is arranged on the modern group system of small tables, each seated for parties of six or eight, and has accommodation for 200 persons. The companion way leads in two flights to the bridge deck, and thence to the promenade deck, where are situated the music room, lounge, library and smoking room. The music room, like the dining saloon, is

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treated in the Georgian style, but finished in lightly-stained mahogany below and ivory white above, with richly-figured brocade panels, the upholstery being made to harmonize. There is a large and airy nursery. The second saloon, on the main deck, is arranged to seat over 200 passengers. The cabin accommodation is heated throughout by steam, the pipes being so arranged that each individual passenger can modify the heat to suit himself. The emigrants are kept comfortable by an installation of the thermo-tank system, which combines the duties of heating and ventilation.

The turbines, which are to be supplied by the Parsons Marine Steam Turbine Company, Limited, are expected to prove of ample power to drive the vessel at a high rate of speed, although the space they occupy is considerably less than would have been required by reciprocating engines of equal power, thereby enabling a greatly improved class of accommodation on the upper decks to be provided. Steam at high pressure will be supplied to the turbines by nine single-ended boilers, fitted with forced draft. The boilers and a very complete installation of auxiliary machinery have been supplied from the engine works of the builders, Messrs. Alex. Stephen and Sons, Limited. The *Virginian* is intended to sail on April 6 on her first voyage from Liverpool to Halifax and St. John, N. B.—"Engineering," London, February 10, 1905.

An announcement recently made of the trial of the Allan Line Steamer *Victorian*, in which her turbines were reported as having proven a failure, seems to have been premature—neither the *Victorian* nor the sister ship, the *Virginian*, has as yet had a trial, but it is expected that these will take place about the first of March, at which time the advocates of turbines for steamship propulsion anticipate a very favorable report.

A number of orders for vessels are being held back awaiting the result of the trials of these vessels.

Lama, *Launch of, December 8, 1904.—Lama*, steel turbine steamer; 275 by 44 by 25.6 feet. Built to the order of the

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British India Steam Navigation Co.'s Colonial trade. The vessel is of the poop, bridge and forecastle type, having accommodation for first-class passengers in the bridge, for second-class in the poop, and for crew in the forecastle. The first-class passengers are berthed in large, airy cabins fitted with all the conveniences now so common in first-class passenger vessels. The dining saloon, which is capable of accommodating all the passengers at once, is in a Romanesque design, finished in cream enamel, the spandrels between arches being filled in with anaglypta diaper, having a mosaic effect. The second-class passengers are accommodated in large and airy three and four-berthed rooms. The main deck is arranged for the carrying of native passengers. All the passenger accommodation is ventilated by electric fans in addition to natural ventilation. As the ship is intended to maintain a high rate of speed, there is only a limited amount of cargo space. The turbines have been supplied by Messrs. Denny & Co., Dumbarton, along with the boilers and the remainder of propelling machinery. The new steamer is classed with the British Corporation and Lloyd's Registers.—"The Steamship."

The Hamburg-American Line is building not only the two enormous steamers *Kaiserin Auguste Victoria* and *America* for the New York-Hamburg run, but also the following smaller vessels for its other routes: At the Bremen Vulcan Works, the *Rhenania*, *Rhaetia* and *Rugia*, of 5,900 tons each, for the East Asia-Hamburg routes; at Kiel Germania-Werft, the *Borussia*, of 7,500 tons, to be used as an army transport or troopship; at Hamburg, the *Reiherstieg-Schiffswerft*, *Blohm* and *Voss* and the Flensburger Schiffbau-gesellschaft—the three steamers *Polynesia*, *California* and *Columbia*, of 6,050 tons each, for the west coast of South America service; at the Bremerhaven Seebeck and the Howaldt Work at Kiel one steamer each, of 2,000 tons, for the east Asian coasting service; another 2,000-ton liner at the Seebeck for the intercoloniál West Indian runs; another at Hamburg—at the Reiherstieg—of 3,800 tons; at the Germania of Kiel and at Fairfield, Glasgow, the

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large steamers *Wittelsbach* and *Furst Bismarck*, of 8,600 tons each. This makes a total of fifteen new steamers being constructed at the same time, with a total gross register of 117,850 tons, which gives the great line a gross total of 141 ocean-going steamers, whose register is 711,856 gross tons, a mighty fleet to fly one house flag.—"American Syren and Shipping."

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YACHTS.

Narcissus, *Launch of*, by Fairfield Shipbuilding and Engineering Company, December 20, 1904.

Narcissus, twin-screw turbine yacht, 245 by 27.6 by 16.3 feet. Built to the order of Mr. A. E. Miller Mundy, of Shipley Hall, Derby. She has a Thames measurement tonnage of 782 tons, and has been specially designed by the Fairfield Company, and constructed by them in a most substantial manner in conformity with the requirements of Lloyd's highest yacht class. Appearance being one of the first considerations in a vessel of the pleasure fleet, great attention has been paid to the form of the hull as well as to the lines of sheer, the contours of stern and cutwater, and also to the placing of masts and funnel so as to secure the utmost of symmetry and grace. She has been designed with a continuous main or weather deck, with cabin deck below forward and aft, a long deck house covered by a shade deck amidships, and rail-high forecastle forward. On the shade deck are the captain's room and chart room, with navigation bridge above. In the forward part of the deck house on the main deck is a handsome drawing room, lit by large, square plate-glass windows. This apartment measures about 14 feet square, and will be comfortably warmed by an open coal fire. Aft of the drawing room is the main entrance leading to dining saloon and owner's private apartments on the cabin deck, as well as to the drawing room, and a cozy smoking room, measuring about 14 feet by 10 feet. The position of the galley on the weather deck renders it particularly easy of ventilation, and will obviate any chance of smells in the lower parts of the ship devoted to living spaces. The central deck house is broken by a convenient cross passage from which a ladderway ascends to the shade deck. This passage renders communication easy from side to side without having to go

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round the deck house, as is the case in so many large steam yachts.

The engine hatch measures only 14 feet by 8 feet, the small dimensions being rendered possible by the fact that no part of the machinery extends above the main deck. In this casing is the main starting and maneuvering platform. The after portion of the main deck clear of the shelter afforded by the shade deck is occupied by promenade space. On the cabin deck the forward part is taken up by a spacious forecastle fitted up for 18 seamen and firemen. Immediately aft of the crew space are the officers' quarters and mess room, and the owner's private cabin with valet's room. Aft of this is the dining saloon, extending the full width of the vessel, and occupying 17 feet of her length, and to accommodate 12 persons. There are coal bunkers to contain 120 tons of coal amidships, the after hold being occupied by the screw shaft and electric accumulators. The whole of the living quarters are heated by a well-designed hot-water heating system, having a slow-combustion furnace placed in a recess off the main stokehold. This system is in addition to the open fires, and will be very useful in keeping the vessel aired and warmed during lying-up periods. Throughout the vessel electric lighting is fitted. The *Narcissus* carries two steam launches, one of 27 feet and one of 16 feet in length, together with two cutters, one gig and one small dingy.

The vessel, which marks another step in the application of turbine propulsion to vessels of the yachting fleet, is schooner rigged with two pole masts, and has a scroll carving forward representing the flower after which she is named. She will be driven by twin screws, and the propelling machinery consists of two independent Parsons compound steam turbines and one condenser. There is one high-pressure turbine on the port side and one low-pressure turbine on the starboard side of the ship, and a reversing turbine is incorporated with each. These two turbines will each drive a separate shaft with one propeller on each shaft, and the engine room is fitted

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with all modern appliances for ensuring economy of working. The boilers for generating the steam are two in number, of the cylindrical, multitubular marine type, constructed entirely of steel, and adapted for a working pressure of 180 pounds per square inch.—"The Steam Ship."

A turbine steam yacht containing several new and interesting features has been recently constructed for Sir George Newnes, M. P., by Messrs. Swan, Hunter & Wigham Richardson, of Newcastle-on-Tyne, from the designs of Sir William White, formerly naval constructor to the British Admiralty. The experience gained by the owner during numerous cruises in various parts of the world has led to the incorporation of some novel features in this new vessel, which is of 1,260 tons. A fundamental idea in the design is the adoption of moderate speed and the utilization of the relatively large dimensions in the best possible accommodation. According to the contract, the maximum speed is to be 15 knots, and for this speed ample power has been provided. With regard to the turbines, it has been stipulated that there shall be unprecedented economy of coal at cruising speeds, which involves a new departure as compared with other turbine-propelled yachts. Very large bunker capacity has been provided. Although primarily coal is to be used, the bunkers have been built so as to be available for oil fuel, for the use of which the cylindrical boilers can be readily adapted. Electric power is to be used for nearly all auxiliary purposes—steering, cable work, warping, boat hoisting, ventilation and heating.—"Scientific American."

Khedive's Yacht.—The old-fashioned engines in the Khedive's steam yacht *Mahroussa* will be replaced by turbine machinery. This vessel, which will have the distinction of being the first royal yacht to be fitted with the new engines, is one of the largest yachts afloat, being 400 feet long, 42 feet beam, and 26 feet 5 inches depth. The reconstruction and alterations to the hull will be carried out by A. & J. Inglis, of Glasgow.—"The Nautical Gazette."

Gasoline Launch.—The Gas Engine & Power Co. and Charles L. Seabury & Co. have secured from Mr. Drexel, of Philadel-

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phia, a contract to build a high-speed launch, to be fitted with two gasoline engines, each having six cylinders.

The boat will have a total length of 62 feet, is fitted with twin screws, and designed for a speed of 26 knots.

Motor Boat.—There is building at the Napier Works, in England, a new motor boat, on radical lines, for Mr. Leopold Rothschild. It is expected that she will be very speedy. The skin is to be four-ply cedar and mahogany, with a fine silk lining between, dressed with an elastic preparation. The several thicknesses are sewn together with phosphor-bronze thread. The motor is to have six cylinders, mounted in singles, with large exhaust valves on the port side and equally ample inlet valves on the starboard. The engine, together with all accessory gear, will be installed very low in the boat, to lower the center of gravity of the weights. In front of the large engine there will be a small, single-cylinder starting motor, which drives by means of friction and ratchet clutches on the geared flywheel. All the reversing gears, thurst bearings, etc., will be contained in an oil and watertight casing, and run in submerged oil. Yarrow will make the propellers, and extensive experiments will be carried out.—Extract from "Marine Review."

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OBITUARY.

RALPH ASTON, REAR ADMIRAL (Retired).

Born in Connecticut in 1841; descended from Puritan stock; educated at Chase's Seminary. He spent two years' apprenticeship in a machine shop; and was a fair draughtsman and designer before he entered the Naval Service. He entered the Navy in 1861 as a Third Assistant Engineer, and was ordered to the *Cayuga*, a "ninety day" gunboat, which took part in the battles of the lower Mississippi River, passing Forts St. Philip and Jackson, and taking an active part in the destruction of Confederate vessels which were encountered above those forts, and in the attack on Vicksburg. He was in the battle of Mobile Bay, Grant's Pass and Grand Gulf. Mr. Aston was promoted to a Second Assistant Engineer in 1863, and to a First Assistant Engineer in 1866; served on board the *Ashuelot* in 1867, '68 and '69.

In 1869 he was transferred to the *Terror*, on the North Atlantic Station. He served on board the *Omaha* in the Pacific squadrons, 1871, '72 and '73; Naval Rendezvous, New York, 1875 and '76; experimental duty, 1883 to 1885, on board the *Pensacola* and *Kearsarge;* European Station, 1885 and '86; on board the *Ossipee* in North Atlantic squadron, 1886 and '87; he was promoted to a Chief Engineer in 1888, when he served on experimental duty at New York; he served on board the *Bennington* from 1891 to 1894. He was Chief Inspector at the Continental Iron Works, New York, from 1894 to 1897, when he was ordered to the *Cincinnati*, and served on board that vessel during the war with Spain. In October, 1898, he was transferred from the *Cincinnati* to the flagship *Brooklyn*, where he remained until he was promoted to the rank of Captain, when his sea duties terminated. He then resumed inspection duty in New York, where he remained until he was

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retired, in January, 1902. His retirement was with the rank of Rear Admiral. His total sea service was 21 years and 3 months.

Rear Admiral Aston was an engineer of a high order, combining the scientific and practical qualities to a high degree. He was a modest, unassuming man, but fearless and indefatigable. His good qualities won for him the esteem of every one he came in contact with, and he justly merited that epithet, endearing to seafaring men, of being a *good ship mate.*

He was an indulgent husband and a kind father.

He died in his home in Brooklyn on the 12th of December, 1904, of cerebral hemorrhage. His death was painless.

He was a member of the Military Order of the Loyal Legion; of the American Society of Naval Engineers, and of the Society of Freemasons. B.

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BOOKS RECEIVED.

THE NAVAL CONSTRUCTOR.—SIMPSON.—A very complete and well-arranged hand-book on Naval Architecture. By GEORGE SIMPSON, M. I. N. A. Published by D. VAN NOSTRAND COMPANY.

The book contains many new cuts and tables of useful information, and is especially good in its description of the various standard fittings of ships.

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ASSOCIATION NOTES.

The regular annual meeting of the Society was held at the Navy Department, Washington, D. C., on December 24, 1904, for the purpose of electing officers for 1905.

A count of the votes received resulting in the following officers being elected:

President, Commander A. F. Dixon, U. S. Navy.

Secretary-Treasurer, Lieutenant Commander W. W. White, U. S. Navy.

Members of the Council, Captain George W. Baird, U. S. Navy; Commander J. K. Barton, U. S. Navy; Commander W. M. Parks, U. S. Navy.

Rear Admiral W. L. Capps, U. S. Navy, received a sufficient number of votes to elect, but he having requested that his name be withdrawn before the election, it was decided that the nominee receiving the next higher number of votes should be elected.

SPECIAL MEETING OF COUNCIL.

At a special meeting of the Council, held on January 7, 1905, Lieutenant Commander W. W. White, U. S. Navy, tendered his resignation as Secretary-Treasurer, and Lieutenant Charles K. Mallory, U. S. Navy, retired, was elected to fill the vacancy thus created.

The following statement was presented by the retiring Secretary-Treasurer:

Received, 1904:		
Dues and subscriptions	$3,875.40	
Advertisements	1,413.13	
Sales, JOURNALS and reprints	1,114.25	
Total receipts 1904	$6,402.78	
Balance received January 1, 1904	5,939.34	
	$12,342.12	12,342.12
Interest on deposits in bank		120.34
		$12,462.46

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Expended:		
Printing JOURNALS	$4,699.89	
Engraving, cuts, etc	262.73	
Underwood typewriter	85.00	
Prize essay and medal	105.00	
Purchase of JOURNALS	305.50	
Lithographs, special	24.00	
Purchase of British Reports	4.41	
Check replaced and payment adjusted	6.32	
Incidentals, postage, telegrams, etc	71.27	
	$5,564.12	
Salary Secretary-Treasurer, 1904	900.00	
	$6,464.12	6,464.12
Credit balance January 1, 1905		$5,998.34

This statement was audited by a committee of the Council and found to be correct.

CHAS. K. MALLORY, *Lieut., U. S. N., Retired,*
Secretary-Treasurer.

A. F. DIXON, *Commander, U. S. N.,*
President.

The Council decided to have prepared a complete index of the JOURNAL, Volumes I to XVI inclusive. Notice will be given when this is ready for issue.

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BY-LAWS OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

1. The Association shall be known as the AMERICAN SOCIETY OF NAVAL ENGINEERS.

2. The object of the Society shall be to promote a knowledge of marine engineering and naval architecture by reading, discussing and publishing papers on professional subjects; by bringing together the results of experience acquired by engineers in all parts of the world, which, though valueless when unconnected, tend much to the advancement of engineering when published together in the JOURNAL of the Society; by publishing the results of such experimental and other inquiries as may be deemed essential to the advancement of the science; and historical events in the lives of engineers.

3. The officers of the Society shall be a President, a Secretary-Treasurer, and a Council, all of whom shall be elected annually.

4. The Society shall be composed of Members, Associates and Honorary Members.

5. Officers of the Line and Construction Corps, and ex-officers of the Engineer, Line and Construction Corps of the Navy, and officers and ex-officers of the Revenue Cutter Service shall be eligible as Members.

6. Persons in civil life whose knowledge of engineering is such that they can co-operate with naval engineers in the promotion of professional knowledge, or who are intimately connected with the engineering profession, shall be eligible as Associates.

7. The Secretary of the Navy, the Assistant Secretary of the Navy, the Chief of the Bureau of Steam Engineering of the Navy Department, all ex-Chiefs of the Bureau of Steam Engineering, the prize essayist of each year, and such other persons as the Society may elect, shall be Honorary Members.

8. Members shall be admitted upon application and payment of the annual subscription.

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9. An Associate may be admitted upon application in writing and payment of the annual subscription, provided his application have the recommendation of a member, and receive the approval of a majority of the council.

10. Associates shall be entitled to all the privileges of Members except voting and holding office.

11. To be eligible as Honorary Member the candidate must receive the unanimous vote of the Council before his name shall be presented to the Society; and the favorable vote of two-thirds of the Members voting shall be necessary for election.

12. The direction and management of the affairs of the Society, and the editing and publishing of the JOURNAL, shall be vested in a Council composed of five members, the President and the Secretary-Treasurer being members *ex officio.* The Council may appoint one of its number Librarian.

13. The President shall exercise the usual duties of that office.

14. The Secretary-Treasurer chall conduct the correspondence of the Society and its financial transactions. He shall submit an annual statement of the receipts and expenditures of the Society, which shall be audited by three other members of the Council.

15. The Council shall have authority to fill vacancies which may occur during the year.

16. Subscriptions and all matter intended for publication, shall be sent to the Secretary-Treasurer; but no paper shall be read before the Society at the annual meeting, nor anything published in the JOURNAL, without the approval of the Council.

17. In deciding matters pertaining to the Society, none but members shall be entitled to vote; to constitute a quorum for business, nine members must be present.

18. The Society shall meet annually in the City of Washington, and at such other times and places as meetings can be conveniently arranged.

19. The annual subscription shall be five dollars, payable in advance.

20. Copies of the JOURNAL may be sold by the Society at a price which shall be fixed by the Council.

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21. Absent members voting on any proposition must state over their signatures whether they are for or against the proposition.

22. On the first Tuesday in October a meeting of the Society shall be held for the consideration of nominations to offices falling vacant under the rules. Nominations to office must be made in writing, and may be made by any member, whether present at the meeting or not. The consent of the nominee to accept office if elected must be assured. A necessary qualification of all nominees, besides full membership, is residence in or near Washington during the term of office.

23. After the nominations are all made, lists of the candidates for the different offices shall be prepared by the Secretary-Treasurer, and mailed to all members of the Society. Each voting list must be signed, and must show under the different offices the name of the person voted for. Each voter will enclose his ballot in a sealed envelope, which will be sent to the Secretary-Treasurer previous to the annual meeting. At the annual meeting the envelopes will be opened and the votes counted.

24. Nothing in the preceding section shall preclude the name of any candidate from being put in nomination for more than one office. In case of a tie, the presiding officer shall decide. Voting by proxy is prohibited.

25. In the event of a candidate receiving the greatest number of votes for more than one office, he shall decide which he will accept.

26. Any office falling vacant in the course of the year owing to the death, resignation or removal from Washington or vicinity of the incumbent, shall be filled by vote of the members of the Council remaining.

27. Motions to amend these By-laws, or to make new By-laws, must be in writing, and must lie over at least three months, during which time notice thereof shall be sent to all members; and the assent of two-thirds of the members voting shall be necessary to amend or adopt.

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AMERICAN SOCIETY OF NAVAL ENGINEERS—1905.

OFFICERS FOR 1905.

President:

Commander A. F. Dixon, U. S. Navy.

Secretary and Treasurer:

Lieutenant Chas. K. Mallory, U. S. Navy (retired).

Council:

Captain George W. Baird, U. S. N.
Commander A. F. Dixon, U. S. N.
Commander J. K. Barton, U. S. N.
Commander W. M. Parks, U. S. N.
Lieutenant Chas. K. Mallory, U. S. N. (retired).

HONORARY MEMBERS.

The Secretary of the Navy.....Navy Department, Washington, D. C.
The Assistant Secretary of the Navy.....Navy Department, Washington, D. C.
Charles Whiteside Rae, Rear Admiral, U. S. N., Engineer-in-Chief, Navy Department, Washington, D. C.
Chas. H. Haswell, Consulting and Superintending Engineer, 324 West 78th street, New York, N. Y.
B. F. Isherwood, Chief Engineer, U. S. N. (retired).....111 East 36th street, New York, N. Y.
J. W. King, Chief Engineer, U. S. N. (retired).....3231 Powellton avenue, Philadelphia, Pa.
Chas. H. Loring, Chief Engineer, U. S. N. (retired).....239 Clermont avenue, Brooklyn, N. Y.
Wm. H. Shock, Chief Engineer, U. S. N. (retired).....1404 15th street, Washington, D. C.
Geo. W. Melville, Rear Admiral, U. S. N. (retired).....Union League Club, Philadelphia, Pa.
George R. Dunell, Esq .....33 Spencer Road, Chiswick W., England.

Prize Essayists.

W. W. White (1897), Lieutenant Commander, U. S. N.
W. F. Durand (1898).....Sanford University, California.
B. C. Ball (1899).....Willamette I. & S. Works, Portland, Oregon.
W. F. Worthington (1900), Commander, U. S. N.
J. R. Edwards (1901), Commander, U. S. N.
F. W. Bartlett (1902), Lieutenant Commander, U. S. N.
Charles W. Dyson (1903), Lieutenant Commander, U. S. N.

MEMBERS.

Adams, Robert B., Engineer, U. S. R. C. S.....U. S. S. *Algonquin*, San Juan, P. R.
Addicks, W. R., Vice President Consolidated Gas Company...No. 4 Irving Place, New York, N. Y.
Allderdice, W. H., Lieutenant Commander, U. S. N.
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Saito, M., Lieutenant, I. J. N.....Care Vickers Sons & Maxim, Barrow-in-Furness, England.
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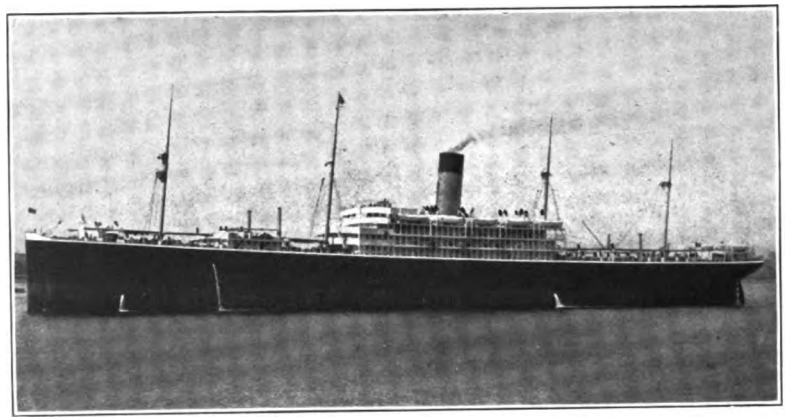

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NEW SCOUT CRUISERS *CHESTER*, *BIRMINGHAM*, *SALEM*.

By W. F. SICARD.

By an Act of Congress, approved April 27, 1904, provision was made for the construction of—

One first-class battleship, of not more than 16,000 tons trial displacement.

Two first-class armored cruisers, of not more than 14,500 tons trial displacement.

Three scout cruisers, of not more than 3,750 tons trial displacement.

All the above vessels to be built under the provisions of the Act of August 3, 1896, as to material for hulls, engines, boilers, machinery, etc., except that no premiums are offered and that all parts shall be of domestic manufacture. The Act states that not more than two of these vessels shall be built by one contracting party, and further authority was given the Secretary of the Navy that he "may build any or all of the vessels herein authorized in such navy yards as he may designate, and shall build any of the vessels herein authorized in such navy yards as he may designate, should it reasonably

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appear that the persons, firms or corporations, or the agents thereof, bidding for the construction of any of said vessels have entered into any combination, agreement or understanding the effect, object or purpose of which is to deprive the Government of fair, open and unrestricted competition in letting contracts for the construction of any of said vessels."

As has been previously noted in the JOURNAL, the contracts for the battleship and the armored cruisers have already been awarded, that for the battleship, No. 25, *New Hampshire*, to the New York Shipbuilding Co., Camden, N. J., those for the armored cruisers, *Nos. 12 and 13*, *North Carolina* and *Montana*, to the Newport News Co.

Regarding the scout cruisers, the Act reads:

"Three scout cruisers of not more than three thousand seven hundred and fifty tons trial displacement, carrying the most powerful ordnance of vessels of their class; to have the highest speed compatible with good cruising qualities and great radius of action, and to cost, exclusive of armament, not exceeding one million eight hundred thousand each."

The interval between the passage of this Act in 1904 and issuing the circulars for the scouts in February, 1905, was employed in perfecting the design of these vessels.

The type of vessel to which the scout cruisers belong is a new one, just being introduced, and the three described here are the first to be built for our Navy. The British are just completing their first vessels of this class, six in number, and are now running off their trials.

Before proceeding with a detailed description of the *Chester* and class, it may be of interest to compare them with the new British scouts. The general particulars are given on the opposite page.

While the table shows that on paper the speed of the American boat is one knot less, it is well to note that her greater size and freeboard should enable her to sustain an average sea speed, in all kinds of weather, at least equal to the British boats, while the difference in the coal capacity gives a vastly superior radius of action.

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	British scouts.			U. S. scouts.
	Adventure. Attentive.	*Foresight. Forward.*	*Pathfinder Patrol.*	*Chester* Class.
Displacement, tons	2,910	2,545	2,500	3,750
Length, feet	374	360	360	423
Beam, feet	38¼	38½	38	46.6
Draught, feet	13¾	13	13½	16¾
I.H.P.	16,000	16,000	17,000	16,000
Armament	10 12-pdrs.	10 12-pdrs.	10 12-pdrs.	12 3-inch.
Torpedo tubes	2	2	2	2
Speed, knots per hour	25	25	25	24
Coal, tons	470	380	380	1,250

The scout cruiser is a very interesting class of vessel, and presents many points of great professional interest to the marine engineer. Being vessels of relatively small displacement and high speed, great power is required to drive them as they approach their limit of speed. The problem, on the other hand, is different from that presented by the "Destroyer" type, in that the scout must be able to sustain high speed indefinitely, requiring more reliable and therefore heavier machinery. The difficulty of installing this powerful machinery on the limited weight available requires that it be very carefully designed and every superfluous pound omitted. The fact that this appropriation provides for three vessels of this high-speed class, affords an ideal opportunity for comparison of different systems of propulsion applied to similar vessels of the same size and type, and installations of reciprocating engines and turbines of different design would give most valuable data and experience to apply in the design of other vessels.

The plans and the specifications issued by the Department called for reciprocating engines; the circular, however, in advertising for these vessels to be built under contract invited proposals for their construction under two classes, as follows:

Class 1.—For the construction of the hull and machinery, including engines, boilers and their appurtenances, and for equipment, complete in all respects, in accordance with the plans and specifications provided by the Secretary of the Navy, and for the installation of ordnance and ordnance outfit.

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Class 2.—For the construction of the hull, and for equipment, complete in all respects, and for the installation of ordnance and ordnance outfit, in general accordance with the plans and specifications provided by the Secretary of the Navy, but on the bidder's design of machinery, preference being given, other things being equal, to a turbine installation.

The specifications submitted by the bidders under Class 2 must cover fully the ground covered by the Department's specifications, and must be so drawn as to cover the detailed requirements of the different Bureaus of the Department. * * * Any design of machinery submitted must be in such detail and accompanied by such plans, specifications and calculations as to enable the Department to readily determine its value, and to ascertain whether it fulfills the general requirements herein mentioned; the calculations to include complete calculations of weights and centers of gravity, and also such calculations as will demonstrate the satisfactory strength of the whole.

The maximum time allowed for completion will be limited by the Department to thirty-two months if reciprocating engines are adopted, and thirty-six months if turbines are adopted, and no bids will be considered which propose to exceed those limits. Failure to complete the vessel within the time specified in the contract will involve penalties as follows, viz: $100 a day for the first month succeeding the expiration of the period fixed by the contract, and $200 a day thereafter until the vessel is completed and delivered as prescribed by the contract.

The requirements as to trials are as follows:

I. A progressive trial over a measured-mile course, for standardizing the screws, extending from maximum speed down to about 12 knots; about sixteen runs to be made over the course in order to cover the range of speed desired, these runs to be made in a depth of water of not less than 30 fathoms.

II. A full-speed trial of four hours' duration in the open sea in deep water, the speed developed by the vessel to be not less than 24 knots an hour, as determined by the revolutions of the main engines. The air pressure not to exceed an average

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of five inches of water, and the steam pressure in the high-pressure valve chests not to exceed 250 pounds. No live steam shall be admitted to the receivers. The full-speed trial displacement not to exceed 3,750 tons.

III. An endurance and coal-consumption trial of twenty-four hours in the open sea, in water not less than 30 fathoms in depth, at a speed as nearly uniform as possible, which shall average for the twenty-four hours not less than 22½ knots, as determined from the corresponding revolutions taken from the standardization trial results. Record shall be made of the coal consumed. All necessary auxiliaries shall be in operation. Samples of the coal used shall be taken during this trial for purposes of analysis and determination of heating value.

IV. An endurance and coal-consumption trial of twenty-four hours in the open sea at an average uniform speed of 12 knots, as nearly as possible. This trial shall be made as nearly as possible under service cruising conditions and the coal consumption for all purposes carefully determined.

There are the usual penalties for over time, over weight and under speed, with provision for acceptance at a reduced price, or for non-acceptance.

The bids were opened at the Navy Department, at noon, April 15, 1905, six firms bidding. No firm can, under the Act, be allowed more than two vessels. The bids were as follows:

SCOUT CRUISER BIDS.

MARYLAND STEEL COMPANY.

One ship, Department's plans, 32 months	$1,875,000
*One ship, turbines, 36 months	1,875,000

NEW YORK SHIPBUILDING COMPANY.

One ship, Department's plans, 32 months	1,824,000
One ship, turbines, 36 months, two shafts	1,830,000
One ship, turbines, 36 months, 3 shafts	1,866,000

Westinghouse turbines proposed.

*No performance guaranteed.

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WILLIAM CRAMP & SONS.

One ship, Department's plans, 32 months..... 1,793,000
Two ships, Department's plans, 31 and 32 months.....each 1,750,000
One ship, turbines, 36 months, 3 or 4 shafts..... 1,885,000
Two ships, turbines, 35 and 36 months, 3 or 4 shafts.....each 1,825,000
Parsons turbines proposed.

BATH IRON WORKS.

One ship, Department's plans, 32 months..... 1,774,000
Two ships, Department's plans, 32 months.....each 1,729,000
One ship, Department's plans, 32 months, 12 boilers..... 1,767,000
Two ships, Department's plans, 32 months, 12 boilers.....each 1,738,000
* One ship, turbines, 36 months, 12 boilers, 3 shafts, cruising turbines..... 1,671,000
* Two ships, turbines, 36 months, 12 boilers, 3 shafts, cruising turbines.....each 1,623,000
* One ship, turbines, 36 months, 12 boilers, 3 shafts, no cruising turbines..... 1,639,000
* Two ships, turbines, 36 months, 12 boilers, 3 shafts, no cruising turbines.....each 1,595,000
One ship, turbines, 36 months, 8 boilers, 3 shafts, cruising turbines..... 1,654,000
Two ships, turbines, 36 months, 8 boilers, 3 shafts, cruising turbines.....each 1,606,000
One ship, turbines, 36 months, 8 boilers, 3 shafts, no cruising turbines..... 1,611,000
Two ships, turbines, 36 months, 8 boilers, 3 shafts, no cruising turbines.....each 1,573,000
* One ship, turbines, 36 months, 12 boilers, 4 shafts, cruising turbines..... 1,688,000
* Two ships, turbines, 36 months, 12 boilers, 4 shafts, cruising turbines.....each 1,640,000
* One ship, turbines, 36 months, 12 boilers, 4 shafts, no cruising turbines..... 1,658,000
* Two ships, turbines, 36 months, 12 boilers, 4 shafts, no cruising turbines.....each 1,610,000
One ship, turbines, 36 months, 8 boilers, 4 shafts, cruising turbines..... 1,671,000
Two ships, turbines, 36 months, 8 boilers, 4 shafts, cruising turbines.....each 1,622,000
One ship, turbines, 36 months, 8 boilers, 4 shafts, no cruising turbines..... 1,640,000
Two ships, turbines, 36 months, 8 boilers, 4 shafts, no cruising turbines.....each 1,592,000
Parson's turbines proposed. Normand boilers.
Bids marked * guarantee 24½ knots.

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UNION IRON WORKS.

One ship, Department's plans, 32 months	1,689,000
Two ships, Department's plans, 32 months.....each	1,666,000

FORE RIVER SHIPBUILDING COMPANY.

One ship, Department's plans, 30 months	1,629,000
Two ships, Department's plans, 30 and 31 months.....each	1,557,000
One ship, turbines, 34 months	1,540,000
Two ships, turbines, 34 and 35 months.....each	1,468,000
One ship, reciprocating engines, 30 months	1,598,000
Two ships, reciprocating engines, 30 and 31 months.....each	1,524,000
One ship, turbines, 34 months	1,572,000
Two ships, turbines, 34 and 35 months.....each	1,500,000
Two ships, { One, turbines; One, reciprocating engines, 34 and 30 months, each.. }	1,525,000
Two ships, { One, turbines; One, Department's plan, 34 and 30 months, each... }	1,556,000

Curtis turbines proposed. Fore River boilers.

Contracts were awarded by the Department as follows:

Scout cruiser *No. 1*, *Chester*, Parsons turbine, to the Bath Iron Works, Bath, Me. Scout cruiser *No. 2*, *Birmingham*, Department's design with reciprocating engines, and *No. 3*, *Salem*, Curtis turbines, to the Fore River Shipbuilding Company, Quincy, Mass.

GENERAL DESCRIPTION.

The vessels and machinery are to be subject to the usual strict Government specifications and rigid inspection.

The general dimensions and features are as follows:

Length between perpendiculars, feet	420
Breadth, molded, feet and inches	46– 8
Displacement on speed-trial, not more than, tons	3,750
Mean draught to bottom of keel at trial displacement, not to exceed feet and inches	16– 9½
Total coal-bunker capacity, about, tons	1,250
Coal carried on speed trial, tons	475
Feed water carried on speed trial, tons	50

The hull is to be of steel throughout.

The armament will consist of twelve 3-inch rapid-fire guns, two submerged torpedo tubes, and a full allowance of rifles, revolvers and cutlasses for the crew.

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The magazines are at each end of the ship, of equal capacity, with ventilated spaces next to the bulkhead, where same separates them from heated compartments.

The ammunition hoists, electrically driven, and capable of supplying each 12 pieces per minute, deliver their ammunition as near as possible to the place where it is required.

The sides of the vessel will be covered throughout the limits of the machinery space with 2-inch nickel-steel, to an average height of about 8 feet above and 3 feet below the water line.

PROPELLING MACHINERY.

The vessel will be propelled by twin screws, turning outboard when going ahead, each screw being driven, through its shafting, by a vertical, inverted, direct-acting, triple-expansion, four-cylinder engine. Each engine will be in a separate watertight compartment, these compartments being arranged, as on a destroyer, one forward of the other. The main engine cylinders are arranged as follows, beginning forward: Low-pressure, high-pressure, intermediate-pressure and low-pressure. The four cranks are at 90 degrees, the forward low-pressure and high-pressure cranks being opposite, as are the intermediate and after low-pressure, the sequence being high-pressure, intermediate-pressure, forward low-pressure and after low-pressure.

The general particulars of engines will be as follows:

Diameter of high-pressure cylinder, inches	28¼
intermediate-pressure cylinder, inches	45
forward low-pressure cylinder, inches	62
after low-pressure cylinder, inches	62
Stroke of all pistons, inches	36
Indicated horsepower of both main engines	16,000
Revolutions	200
Steam pressure, at high-pressure receiver, pounds	250

The engines are of the usual Bureau design, with open framing. The forward low-pressure and high-pressure cylinders are bolted together, as are the intermediate and the after low-pressure cylinders. There are fore-and-aft tie rods between cylinders, and pads for athwartship tie rods in case they are

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found necessary. There is one set of valve gear for each cylinder, interchangeable as far as possible, and all valves are of the piston type with Stephenson link motion. The engines are provided with all the necessary auxiliaries and accessories, in accordance with the latest practice of the Bureau of Steam Engineering.

The main air pumps are independent and double acting, the circulating pumps are centrifugal and each capable of discharging 11,000 gallons of water per minute from the bilge. Steam turbines may be used for the circulating pumps, provided the guaranteed steam consumption does not exceed 50 pounds of steam per indicated horsepower per hour.

Each engine has an independent, steel shell, cylindrical condenser of about 8,000 square feet of cooling surface. In addition to this there is an auxiliary condenser of 600 square feet surface, and a dynamo condenser of 200 square feet surface.

There will be twelve water-tube boilers of the "Express" type, in six water-tight compartments. The steaming capacity will be such that all steam machinery on board can be run at full power with an average air pressure in the ashpits of not more than 5 inches of water. The total grate surface will be at least 693 square feet, and the total heating surface at least 37,080 square feet. The working pressure will be 275 pounds per square inch.

All pressure parts of boilers will be constructed entirely of open-hearth steel plate and seamless steel tubes. No malleable or cast iron or cast steel will be used under pressure. There will be no screw joints in contact with the fire.

The provision for removing and renewing tubes, the method of baffling to provide for the proper circulation of the gases, and the facilities for cleaning soot off tubes and baffles will be fully shown on plans submitted by contractors.

The boiler tubes will be "cold-drawn seamless steel tubes," of the following thicknesses:

1 inch diameter, outside, ...	No. 12, B.W.G.
1⅛ inches diameter, outside, . . .	No. 11, B.W.G.
1¼ inches diameter, outside, . . .	No. 11, B.W.G.

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The steam and water drums will be of the following dimensions:

Steam drums, not less than 40 inches, inside diameter.

Water drums, not less than 18 inches, inside diameter.

All rivets will be in accordance with the Navy standard, and where possible hydraulic riveting will be used; where hydraulic riveting cannot be used, the rivet holes will be coned and conical rivets used. All rivets will be driven with at least 60 tons pressure and held until cool. No snap riveting will be used on any part of the boilers except the casing. Seams will be calked on both sides in an approved manner. If welded drums are used, the weld must be plainly marked, and the tube holes must not be drilled in weld.

Tube holes should be smooth, true and absolutely free from scoring or grooving. The edges of all tube holes in boiler drums must be sufficiently rounded to prevent cutting the tubes when expanded.

The closed fireroom system of forced draft is used, air being supplied by six blowers, located on the deck above the machinery space.

There are four smoke pipes, about 72 feet in height above the base line.

The following auxiliary steam machinery of approved make and design, in addition to that pertaining to the main engines and dependencies, is to be supplied by the contractor and installed complete with all piping and fittings: Steering engine, windlass engine, ash-hoist engines for each fireroom, forced-draft blowers, evaporating plant, to consist of not less than four units, having a total capacity of 16,000 gallons of fresh water per day; a distilling apparatus capable of condensing at least 16,000 gallons of water per day. The workshop tools and the ice machines will be operated by steam engines or electric motors. There will be a refrigerating plant with a cooling effect of 2 tons of ice per 24 hours.

The vessel is to be heated with steam throughout, and the usual steam-heating system will be provided and installed by the contractor.

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The weight of all machinery and tools, stores and spare parts will be about 798 tons. This weight must include all articles, except stores supplied by the Government, irrespective of name or use, coming under the cognizance of the Bureau of Steam Engineering, including water in boilers, condensers, evaporators, tanks, piping, etc., but excluding the reserve feed water in the double bottom or tanks.

The following machine tools, of approved make and design, will be furnished and installed by the contractor in the general workshop:

One screw-cutting back-geared extension gap lathe, to swing 16 inches over the upper ways and 32 inches over the lower ways, and to take between centers 7 feet when extended.

One 12-inch screw-cutting back-geared tool-room lathe, to take not less than 4 feet between centers.

One column tool-room shaper of about 15 inches stroke and not less than 15 inches traverse.

One upright drill press, to drill up to 1½ inches in steel, 14 inches from edge of work, with at least 14 inches traverse of spindle.

One universal milling machine, with not more than 18-inch table feed, 4½-inch traverse and 13-inch vertical feed.

One combined hand punch and shears, with 6-inch shear blades, to cut ¾-inch round iron, shear ⅜-inch steel plate, and punch ⅜-inch holes in ⅜-inch mild steel plates 4 inches from edge.

One emery grinder on column, with two carborundum wheels 12 inches in diameter and 2-inch face.

Three bench vises.

One blacksmith forge with tools and fixtures will be supplied and installed in the blacksmith shop.

Electric Plant.—The vessel will be lighted throughout by electricity. The electric plant will consist of three 32-kilowatt steam-driven generating sets, all to be of 125 volts pressure at the terminals, 600 electric fixtures, complete, with necessary incandescent lamps and outlets; two arc lamps for each engine room, one arc lamp for each firing position in boiler rooms,

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PUMPS AND CONNECTIONS.

No.	Pumps.	Size (inches).	Kind.	Suction pipes from—		Discharge pipes to—		Location.
				Ins.		*Ins.*		
2	Main air....	2 12-inch, 2 22-inch, by 18.	Double, vertical, bucket, double-acting.	11	Condenser.	9½	Main feed tanks.	One in each engine room.
2	Main circulating.	Centrifugal.....	Vertical compound, or turbine.	16½ 6½	Sea. Main drain.	16½ 16½	Condenser. Overboard.	One in each engine room.
2	Main feed.	15 by 10 by 15...	Vertical piston, double-acting, single.	6 6	Feed-tank connecting pipe. Channel ways.	5 5	Main feed discharge main. Main feed tanks.	One in each engine room.
3	Fireroom feed.	9 by 6 by 12.....	Vertical piston, double-acting, single.	2½ 3½ 3½ 3½	Bottom blows in same compartment. *C. and R. pipes from ship's sides. Feed suction pipe. *Reserve feed-water tanks.	3 3 3 2½ 2½	Auxiliary feed. *Main feed tanks. *Reserve feed-water tanks. Overboard in same compartment. Fire extinguisher.	One in each boiler room.
3	Fireroom fire and bilge.	12 by 8 by 12....	Vertical piston, double-acting, single.	4½ 4½ 4½ 1½	Fireroom floors in same compartment. Sea. C. and R. drainage manifolds. Hose connection.	4 4 4 2½	Fire main. Overboard in same compartment. Ash ejector. Hose connection. Fire extinguisher.	One in each boiler room.
2	Fire and bilge.	10 by 8½ by 12...	Vertical piston, double-acting, single.	4½ 4½ 4½	Sea. Engine-room drainage. C. and R. drainage manifolds.	4 4 2½	Fire main. Overboard. Hose connection.	One in each engine room.

Connections marked (*) are on one pump only.

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1	Reserve feed water.	3½ by 4 by 4.....	Vertical piston, double-acting, single.	3	Reserve feed-water tanks.	2½	Feed tank connecting pipe.	Forward engine room.
2	Shaft bilge.	Capacity of each about 300 cubic in. per stroke.	Plunger, single acting.	4	Engine-room bilges.	4	Overboard.	One in each engine room.
2	Distiller circulating.	10 by 10 by 12, or centrifugal.	Horizontal piston, double-acting, single, or turbine.	.....	Sea.	.....	Distillers. Sanitary system. Fire main.	Where directed.
1	Auxiliary air and circulating.	6 by 10 by 10 by 12.	Combined.....	5 5½	Auxiliary condenser. Sea.	3½ 5½	Main feed tanks. Auxil'y condenser.	In forward engine room.
1	Dynamo air and circulating.	6 by 8 by 8 by 12	Combined.....	4 4	Dynamo condenser. Sea.	3 4	Main feed tanks. Dynamo condenser.	In dynamo room.
1	Evaporator feed.	4½ by 5 by 6.....	Light service.....	.....	Sea. Distiller circulating water discharge.	.....	Evaporators.	Where directed.
1	Distiller fresh water.	4½ by 5 by 6.....	Vertical piston, light service.	.....	Bottom of distillers.	.....	Fresh water tanks. Main feed tanks. Reserve feed-water tanks.	Where directed.
2	Engine-room oil.	2 by 1½ by 1¾....	Piston, with hand attachment.	.....	Oil-storage tanks.	.....	Distributing tank, with overflow pipe from tank to pump suction.	One in each engine room.
.....	Other pumps as may be required.		As approved.....	.....	As may be required.	.....	As may be required.	As approved.

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twelve arc lamps for coaling operations, capacity about four ampères; one electric log; two 60-inch searchlights, complete; two double truck lights, one electric night signal set with typewriter form of keyboard; two diving lamps and eight 150-candlepower incandescent lamps; six portable electric ventilating sets, $\frac{1}{4}$-horsepower; thirty $\frac{1}{12}$-horsepower bracket fans; six $\frac{1}{6}$-horsepower bracket fans; one electric whistle operator with automatic and at-will switch; eleven signal lanterns; a set of ship's running-light fixtures and fittings for wireless telegraph outfit; a complete system of interior communication with two dynamotors, batteries and switches for current supply, lights and outlets for interior communication instruments; necessary switchboards and distribution boards for the efficient operation and control of the electric plant, together with the necessary wire supplies, wiring appliances, wiring accessories and instruments for the above and for all other electric auxiliaries and appliances installed; also conduit tubes and fittings, fixtures, lamps, tools, spare parts and stores necessary for the proper manipulation, test, maintenance and repair of the plant.

There will be fitted all the usual means of interior communication, such as telephones, voice pipes, call bells, buzzers, gongs and annunciators, engine and steering telegraphs, revolution and rudder indicators, heeling indicators, automatic fire alarms, warning signals, alarm signals, stoking telegraph, etc.

With the exception of the auxiliaries before mentioned to be operated by steam, all power on board the vessel will be electric, as, for instance, deck winches, ammunition hoists, ventilation blower motors, air compressor motors, etc.

Ventilation.—All main compartments of the ship below the berth deck, except the coal bunkers, will be provided with ventilation, about 30,000 cubic feet of air per minute being required. Special attention will be given to spaces subject to habitually high temperatures, such as engine rooms and dynamo rooms. All blowers for the ventilation system will be electrically operated.

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Coaling Arrangements.—The coal bunkers, of a maximum capacity of 1,250 tons, will be arranged so as to provide a rapid and sufficient supply of coal to the fire rooms. For coaling, four electrically-driven winches will be provided, together with all necessary fixed chutes, scuttles, hatches and other openings. Coaling booms fitted to boat davits will be used in coaling.

Anchor Handling.—Complete arrangements will be provided for the convenient handling and stowage of anchors. There will be one billboard and three hawse pipes, two hawse pipes to be so designed that stockless anchors may be stowed in them. There will be three heavy anchors, one of navy type and two stockless, and the usual smaller ones. Anchor windlass with vertical spindle wildcats, anchor davit, securing and tripping gear, controllers, ring bolts, riding bits, cleats, pad eyes and other fittings will be provided as customary in the naval service.

Boats.—The following boats will be carried, adequate provision being made for their convenient stowage and handling, the necessary boat davits, adjustable boat chocks and all necessary fittings being provided for this purpose:

One 28-foot steam cutter.
One 33-foot launch.
Two 30-foot cutters.
Three 28-foot cutters.
One 30-foot whaleboat.
One 30-foot gig.
One 14-foot dinghy.

Complement.—The arrangement of quarters provides ample accommodations for the following complement: A commanding officer, 12 wardroom officers, 5 warrant officers, not less than 310 men. All wire or metal lockers, bag racks, ditty-box racks, hammock hooks and miscellaneous fittings required for the complete berthing and messing of the crew, and staterooms, mess rooms, etc., for the accommodation of the officers will be provided.

Water Systems.—A complete drainage and flooding system

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will be provided, consisting of auxiliary drain in engine and boiler rooms, drains for all bilges, double-bottom drainage for all compartments within watertight longitudinals and flooding for all compartments within machinery spaces and for trimming tanks. Sounding tubes will be fitted to all of the above double-bottom compartments and trimming tanks, and air escapes provided for such compartments as are to be flooded. There will also be installed magazine floods and a complete fire system, with about twenty fire plugs; a flushing and salt-water system for ready supply of salt water to all parts of the ship; also a fresh-water system for furnishing fresh water to all spaces required. The total quantity of fresh water carried in ship's tanks will be about 6,000 gallons, and in gravity tanks 1,000 gallons.

Plumbing.—Plumbing fixtures will be in accordance with the latest approved practice for all bathrooms, lavatories, water-closets and other spaces.

Means of handling and navigating the ship-steering appliances and other arrangements will be such as have been found most satisfactory on naval vessels. Steering station, with power steering wheels, will be located on the bridge, in the chart house and in steering-engine room. Hand-steering gear will be provided in the steering-engine room.

There will be a bridge forward. On the bridge and in the chart house will be fitted a steering wheel, engine telegraphs, indicators, voice tubes, telephones, and all fittings for handling the ship. There will be a brass chart house, a wireless telegraph room and searchlight platforms suitably located.

Masts.—There will be metal masts forward and aft, the foremast in wake of the chart house to be of brass; both masts to be fitted with wood topmasts. There will be a signal yard fitted on each mast. Foremast will be arranged for wireless telegraphy.

Wood Decks.—Exposed main deck, forecastle deck, top of wireless telegraph house and bridge will be planked; all planking to be of yellow pine, with teak margins.

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Deck Covering and Joiner Work.—Floor coverings of linoleum, tiling or wood gratings will be fitted. The quantity of wood work will be reduced to the minimum, and no wood work will be fire-proofed.

Sheathing.—All spaces in officers' and crews' quarters on gun and main decks, bounded by the outer hull, also sick bay and dispensary, orlop berth deck, also engine-hatch trunks in wake of staterooms, will be sheathed with sheet metal, cork painted. Metal sheathing will also be fitted over the engine rooms and around fireroom hatches on berth deck.

Provision will be made for carrying three months' allowance of provisions and six months' allowance of clothing and small stores, or such portion thereof as may be practicable.

The following compartments and spaces have been provided for and will be fitted out complete as customary in the naval service: Captain's cabins, officers' staterooms, chart house, wardroom country, sick bay, dispensary, berthing spaces, chief petty officers' quarters, commissioned and warrant officers' mess rooms, captain's office, executive officer's office, paymaster's office, engineer's office, prisons, officers' and sick bay lavatories and bathrooms; firemen's and chief petty officers' wash rooms, chief petty officers' water closets, seamen's head, galleys, general mess pantry, pantries, cold-storage rooms, submerged torpedo room, magazines, wireless-telegraph room, galley issuing room, commissary store, magazines, armory, machinery spaces and evaporator room, dynamo room, steering compartment, windlass-engine space, ice-machine room, general workshop, storerooms of all descriptions, rooms for sails and awnings, paymaster's issuing rooms, oilskin lockers, lucky bag, drying room, paint and paint-mixing rooms, lamp rooms, oil rooms, cleaning-gear locker, chain lockers, fresh-water tank spaces, coal bunkers, double-bottom compartments, trimming tanks, reserve feed water tanks.

The following is a summary of the weights to be carried:

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	Tons.
Armament and two-thirds of ammunition and stores,	120
Total protection, nickel-steel, about	183.20
Steam engineering complete, with water in boilers, condensers, piping, etc., and stores, except those furnished by the Government, etc., ...	798
Engineer's stores supplied by the Government, two-thirds full supply, ...	8
Reserve feed water for steaming purposes, . . .	50
Equipment complete, including anchors, chains, electrical plant, etc., and two-thirds full supply of equipment stores, ...	123.2
Coal, trial supply, ...	475
Outfit, including boats, ladders, furniture and blocks,	35
Miscellaneous stores and fresh water, two-thirds full supply, ...	39.90
Officers, crew and effects, ...	45.70
Provisions, clothing and small stores, ...	40

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A DESTROYER IN SERVICE.

ONE YEAR'S PERFORMANCE.

By Lieutenant G. C. Davison, U. S. N., Member.

One of the most valuable features of the Journal is its description of new vessels and accounts of their acceptance trials. But it is a well-known fact that in service trial-trip data is seldom duplicated, owing to various causes, such as increased displacement, foul bottom, deterioration of machinery plant and the element of personnel. It might therefore be interesting and valuable to have trial-trip data supplemented by data collected during actual service. The difference between trial-trip and service performance is probably greater in torpedo craft than in any other class of vessel, for the reason that there is a greater proportionate increase of displacement, and they are more sensitive to any change in the condition of their bottoms.

The *Paul Jones* is the destroyer for which service data is given below. A description of this vessel, and her two sister craft, the *Perry* and *Preble*, with an account of their official trials, is given in Journal No. 3, Vol. XIV. For convenience of reference the following general data is repeated:

Length, feet and inches	250– 7⅜
Beam, extreme, feet and inches	23– 6¼
Mean draught, feet and inches	7– 1½
Displacement on trial, tons	474.5
R.P.M., main engines	322.4
Speed, knots per hour	27.4
Slip, per cent	18.2
Total H.P.	6,925
Coal per H.P. hour, pounds	2.34

On July 19, 1902, the *Paul Jones* was delivered to the Government, and was placed in reserve at the Navy Yard, Mare Island, Cal., where she was kept until December 22, 1903.

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During this period in reserve she was given occasional short runs, of which there is no data available. It is probable, however, that there was little wear of the machinery during this period, and the deterioration only what is normal on all vessels during the same length of time.

She was, therefore, a comparatively new vessel on going into commission on December 22, 1903. From that date data has been observed and recorded every time she has been under way. Several thousand miles have been covered on short runs of from twenty-five to a hundred or so miles, but all such data is not recorded in the accompanying table, as it would be a useless repetition, and in some cases the coal consumption is not accurate. The longer runs are all given, and in general the observations are quite accurate.

This table really tells the whole story of the performance of the *Paul Jones* for a period of a year and a few months. It would, however, be interesting only to officers with past, present and future duty on a vessel of the same class. But an analysis of the table establishes a few facts of general interest and corrects some apparently false impressions, which, while not general in the service are held by some officers. Some of these erroneous impressions are:

(1) That the economical speed is about 14 knots.

(2) That the law of horsepower varying as the cube of the speed does not apply practically.

(3) That destroyers are weak things which have to be nursed to a certain extent and cannot take care of themselves, and in general are not to be relied upon.

It is not surprising that such false impressions exist, as our destroyers are comparatively recent acquisitions, and inherited much of the reputations of their forerunners, the torpedo boats. While destroyers are actually only big torpedo boats, their bigness lifts them into another class of vessel.

To begin with the analysis of service performance, Fig. 1 gives three horsepower and revolution curves. Curve No. I is from data taken with an absolutely clean bottom. The horsepower and revolutions were taken as accurately as indicators

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S. TORPEDO

		Cut offs.							
		Starboard.				Port.			
		H.	M.	1 L.	2 L.	H.	M.	1 L.	2 L.
		·5	·5	·5	·5	·5	·5	·5	·5
	Welsh.	·5	·5	·5	·5	·5	·5	·5	·5
		·4	·6	·8	·8	·4	·6	·8	·8
with barnacles.	and Welsh.	·4	·6	·8	·8	·4	·6	·8	·8
with barnacles.		·4	·6	·8	·8	·4	·6	·8	·8
with barnacles.		·4	·6	·8	·8	·4	·6	·8	·8
with barnacles.		...	...	...	...	...	...	...	...
with barnacles.		·5	·6	·7	·7	·5	·6	·7	·7
with barnacles.		·5	·6	·7	·7	·5	·6	·7	·7
with barnacles.	.....	...	...	...	...	...	...	...	...
l.		·5	·6	·7	·7	·5	·6	·7	·7
l.		·5	·6	·7	·7	·5	·6	·7	·7
l.		·4	·4	·4	·4	·4	·4	·4	·4
		·4	·4	·4	·4	·5	·5	·5	·5
		·4	·4	·4	·4	·5	·5	·5	·5
		·4	·4	·4	·4	·4	·4	·4	·4
		·4	·4	·4	·4	·4	·4	·4	·4
		·4	·4	·4	·4	·4	·4	·4	·4
	Nanaimo.	·4	·4	·4	·4	·4	·4	·4	·4
	Nanaimo.	·4	·4	·4	·4	·4	·4	·4	·4
		·4	·4	·4	·4	·4	·4	·4	·4
		·4	·4	·4	·4	·4	·4	·4	·4
.....	.....	...	...	...	...	...	...	...	...
.....	.....	...	...	...	...	...	...	...	...
ly foul.		·4	·4	·4	·4	·4	·4	·4	·4
ly foul.		·4	·4	·4	·4	·4	·4	·4	·4
ly foul.		·4	·4	·4	·4	·4	·4	·4	·4
ly foul.	.....	·4	·5	·4	·4	·5	·5	·5	·5
.....	.....	...	...	...	...	...	...	...	...
ly foul.		·7	·7	·7	·7	.10	.10	.10	.10
.....	.....	...	...	...	...	...	...	...	...
ly foul.		·4	·4	·4	·4	·4	·4	·4	·4

ained, 330 R. P. M.

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d

cbstrtrtr

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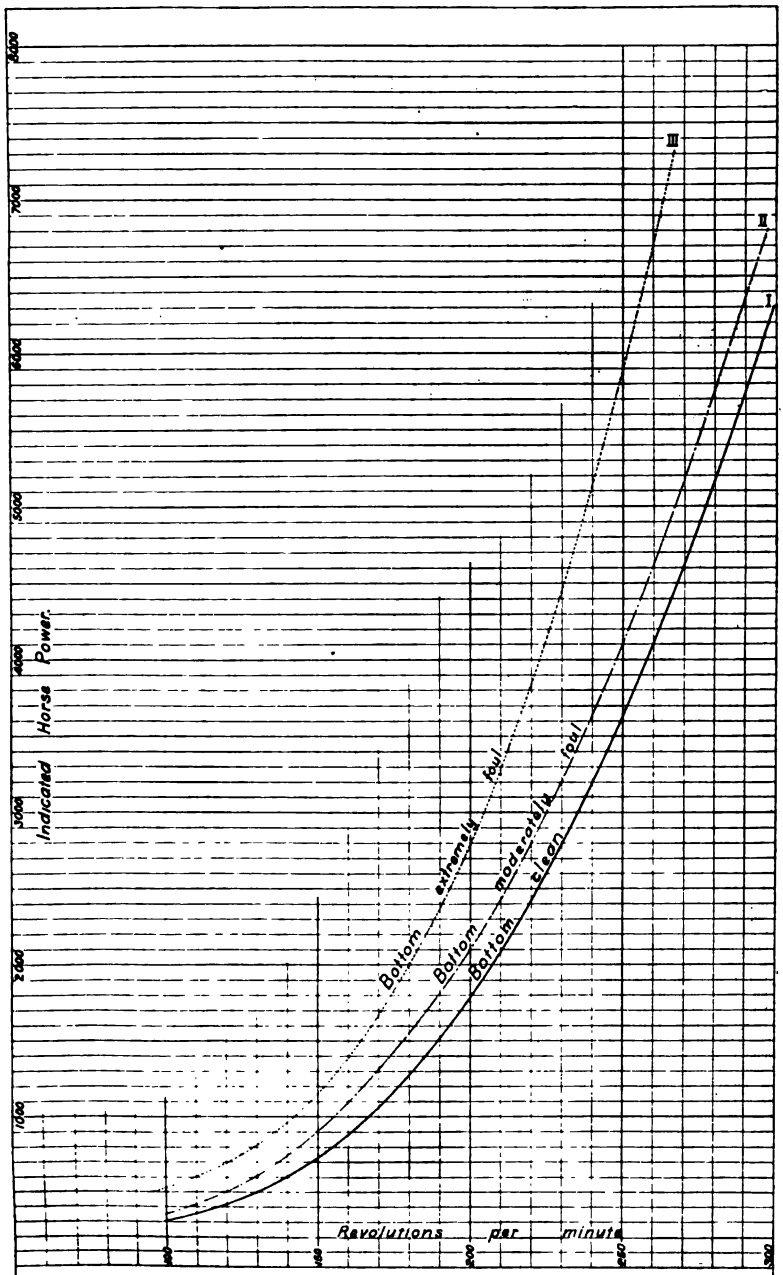

Fig. I.

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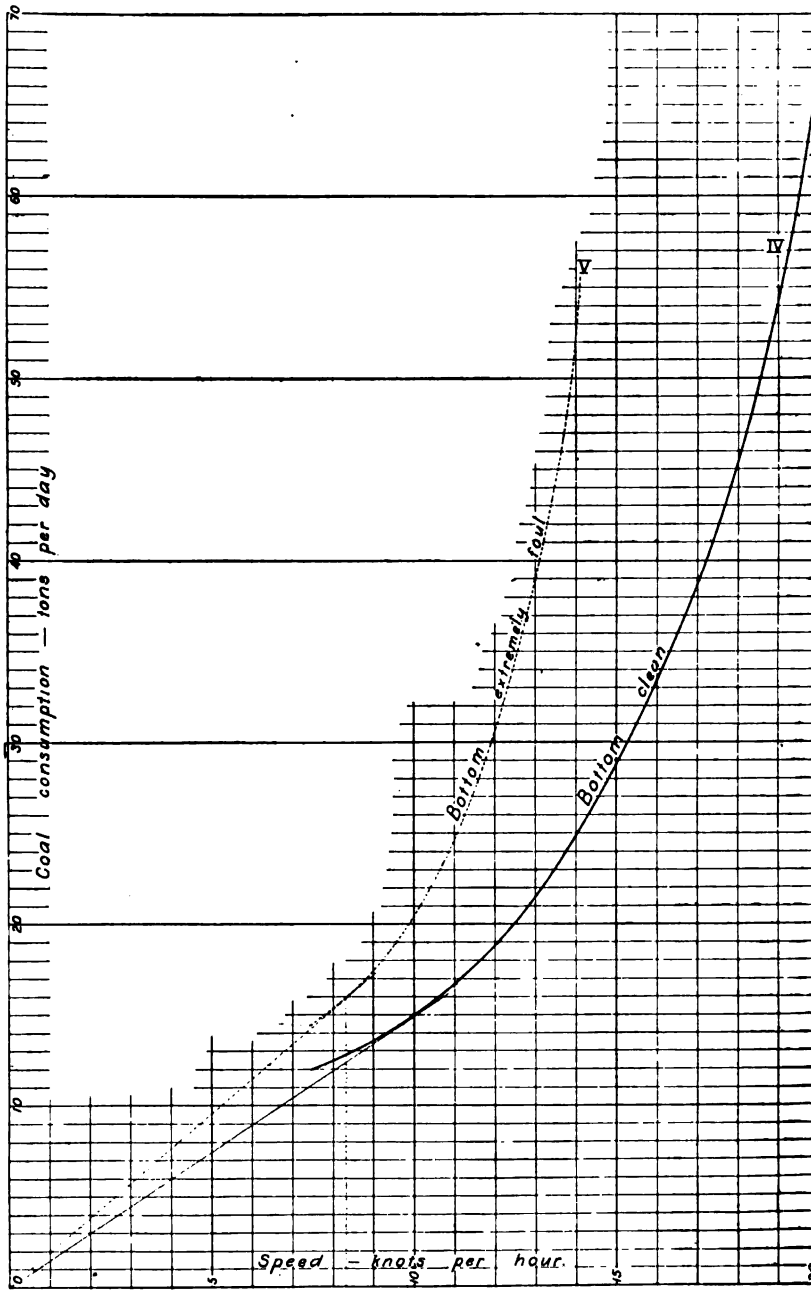

Fig. 2.

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and counters would give them. This curve coincides exactly with a theoretical curve where $HP. = C\,(\text{R.P.M.})^3$ the value of C of course being obtained by substituting horsepower and revolutions as actually observed.

Curve II is made from observations when the bottom was moderately foul, and in the same condition. This condition was made possible by the fact that the vessel was in warm water for two months soon after coming out of dry dock. During these two months she grew a crop of barnacles, about ¼ to ½ inch in height. Then she went to Puget Sound where the water was cold (about 50 degrees Fahrenheit). There barnacles ceased to grow, and no difference could be detected in the condition of her bottom from November, 1904, to April, 1905.

Curve III is made from observations with a very foul bottom. In this case also the foul condition was constant, being caused by a growth of large barnacles 2 to 3 inches in height. After leaving Panama, the barnacles did not grow perceptibly. Two weeks were spent in fresh water, the Columbia River, Oregon, which effectually killed the barnacles. But they did not drop off, and remained attached to the bottom until the vessel was next docked. It seemed that in this condition the bottom was as foul as it could get, so far as any effect on the speed of the vessel was concerned. It will be seen from Curve III that with a very foul bottom the horsepower required for any speed is about *double* that required for the same speed with a clean bottom. This, of course, takes into consideration the increased slip due to foul bottom. It also shows that the maximum speed attainable with a foul bottom is 23 knots as compared to 29 knots with a clean bottom.

It would seem that curves Nos. I and III may be taken as the two extremes. In any intermediate condition of the bottom one accurately determined point will be sufficient to establish an approximate curve for that condition of the bottom, using the law of the cube of the speed. This has actually been done in practice on this vessel, and has given results

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which coincide with actual observations. Such curves are of great value when it comes to estimating the coal required for a certain length of run, the highest speed permissible with the coal supply on board, and finally as a check on the manner of firing. The close agreement of observed points with the theoretical curve based on the cube of the speed law seems sufficient evidence to establish the fact that that law holds good for practical purposes. It seems scarcely necessary to invite attention to the fact that most of the resistance of a vessel of this class is due to a skin friction when her bottom is foul. If it is assumed that with a clean bottom the wave-making and skin resistance are nearly equal, then, with a foul bottom, where the total resistance is doubled, it is evident that the skin friction is about three times as great as the wave-making resistance.

Finally, while the cube of the speed rule does not apply to all vessels, and while theoretically it may not apply exactly to this one, practically it applies so well as to be very useful, which is all that is required in practice.

COAL CONSUMPTION AND ECONOMY.

When in port with steam up during the entire twenty-four hours, the coal consumption averages 3 tons per day. It never exceeds 3½ tons. Generally fires are banked except during six hours of the twenty-four, so that the coal consumption is only 1½ tons per day. Fig. 2, Curve IV, represents coal consumption at sea. Curve IV is the coal-consumption curve with a foul bottom. It is seen that the economical speed with a clean bottom should be 10 knots and with a foul bottom 8. In actual practice these speeds have been used, as this vessel was cruising in company with other vessels which could steam no faster.

Fig. 3 gives curves showing the effect of increased displacement on the slip. Revolutions per mile are used instead of percentage of slip, as it is often of practical value for navigation and other purposes to know the revolutions per mile. Curve VI was made by a series of comparisons between the

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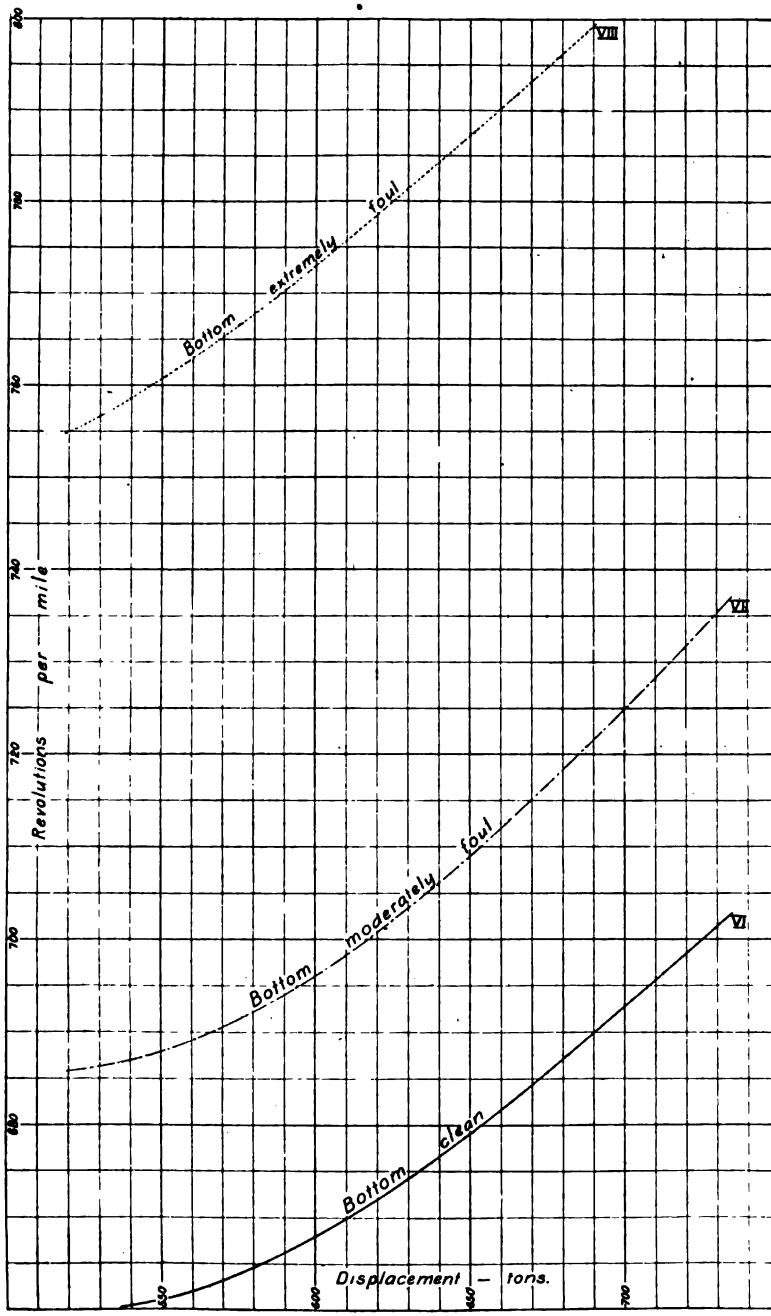

Fig. 3.

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revolution counters and the patent log, thus eliminating current. The effect of wind and sea was eliminated by using data observed in comparatively smooth weather.

That a destroyer can be self sustaining has been demonstrated on a number of occasions for periods of several months. With a complete outfit of stores it would be possible for this vessel to perform active service for a period of six months without going near another vessel or navy yard. The only stores she would have to get during that period are coal and oil. The evaporating plant is capable of supplying all fresh water needed at all times.

On account of the large number of bearings the oil consumption seems large, twenty gallons per day being the least ever used at ten knots. But as much as 1,000 gallons have been carried at one time, so that the oil supply can be made to last fifty steaming days.

The highest speed ever attained on this vessel during this commission was 27 knots. Her bottom was somewhat foul at the time, and, although indicator cards were not taken, the revolutions and receiver pressure indicated that she was developing her maximum horsepower. It would not be far wrong to estimate her maximum speed under average conditions of foul bottom and displacement at 26 knots. As previously shown, it may at times, with an extremely foul bottom, be as low as 23 knots. But even with the above limitations it is believed that these destroyers are today the best in the world. It appears that a happy medium as regards length, beam, displacement, power and strength of hull was struck when these vessels were designed, making them superior to the latest British destroyers of approximately the same size and of only 25 knots trial speed. What speed will the latter make under service conditions with a foul bottom?

To sum up, the destroyers of which the *Paul Jones* is taken as a representative of the type, have the following capabilities in actual service:

1. A cruising radius of 3,300 miles, at ten knots, with a clean bottom.

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2. A cruising radius of 2,700 miles, at eight knots, with a foul bottom.

3. A maximum speed of about 28 knots, with a clean bottom.

4. A maximum speed of about 23 knots, with a foul bottom.

5. Ability to keep at sea in all weather.

6. Comfortable quarters for officers and men.

7. Self-sustaining power for a period of several months.

Thus it is seen there is but a narrow margin between the maximum speed of the fastest heavy ships under favorable, and the maximum speed of a destroyer under adverse conditions. So, if we are to have any more destroyers, it is hoped that their acceptance trial speed will not be less than twenty-nine knots.

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TEST OF PIPE JOINTS.

The following description of tests, made at the works of Dienelt & Eisenhardt, Philadelphia, is taken from a report made by Commander T. F. Burgdorff, U. S. N., member.

The tests consisted in expanding, flaring and facing a 20-inch lap-welded steel pipe, ⅜-inch thick, into a grooved steel flange by a Lovekin pipe-flanging machine, and in testing various forms of flanges of 4- and 5-inch pipe by hydraulic pressure for tightness between flange and pipe, and for determining the force necessary to detach the flanges from their pipes. These tests were conducted at the above named works in my presence on February 14th and 16th, 1905.

The Lovekin pipe-flanging machine, by which the 20-inch steel pipe was secured to its steel flange, resembles a large lathe in appearance. It is about 35 feet 6 inches long, 5 feet 7 inches wide, and about 7 feet high; weighs, approximately, with the set of tool heads, 20 tons, and is capable of expanding into flanges pipes varying from 12 to 30 inches in diameter and in lengths up to 20 feet.

It consists essentially of three parts, securely bolted together, viz:

1st. A heavy head piece, with hinged cap, by which the pipe flange is securely clamped in accurately-fitting flange dies, to prevent the flange from expanding with the pipe. These flange dies are made in half sections, of a uniform outside diameter to fit the head piece, and of an inside diameter to fit the circumference of each size of pipe flange. To fit these flange dies accurately the outer surfaces of the pipe flanges are turned true. This part is also provided with a gripping device to prevent the pipe from turning. A view of the head is seen in the middle part of Fig. 1, which is taken from the "The Iron Age," of February 16, 1905.

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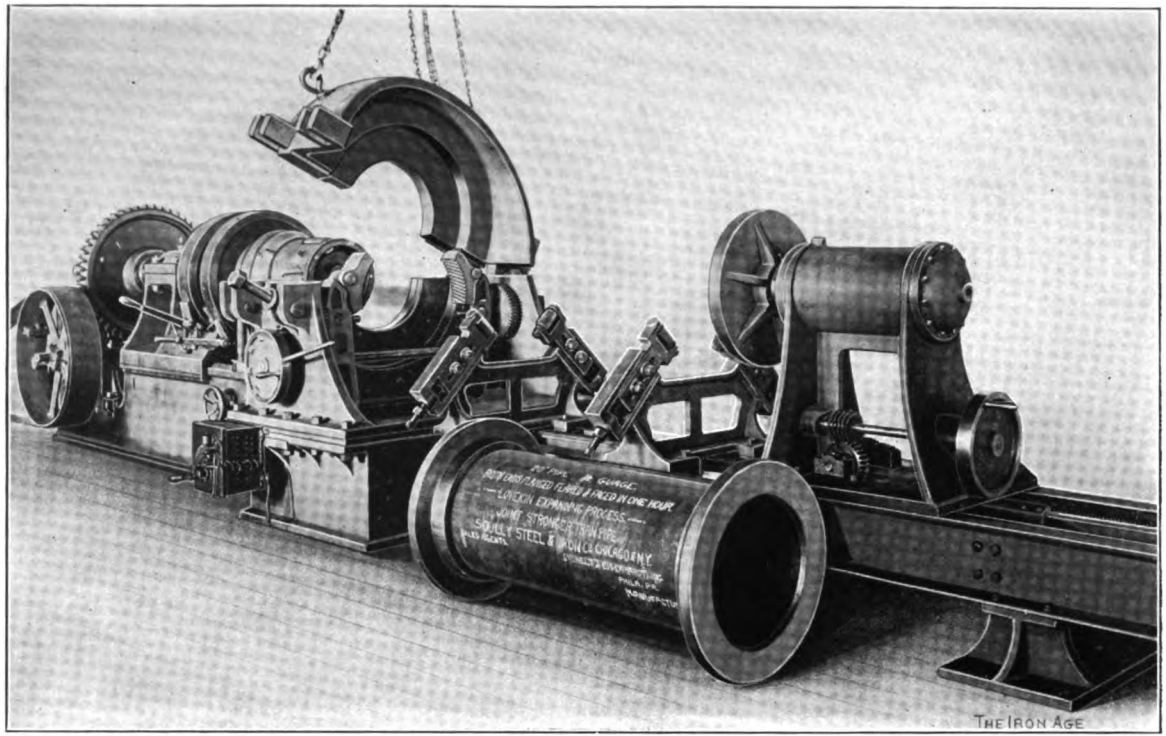

Fig. 1.

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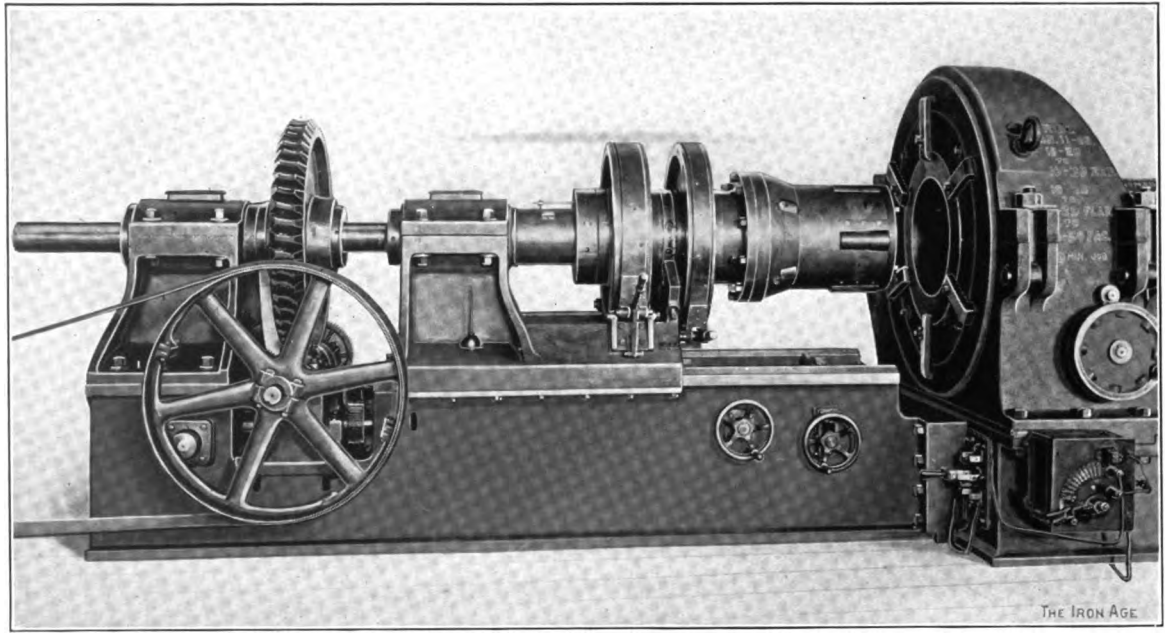

Fig. 2.

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2d. A two-channel beamway fitted with central rack, on which the pipe is supported by means of adjustable roller blocks, and which also supports an adjustable hydraulic ram, by means of which the pipe may be forced into its flange and kept from backing out while being expanded. A view of this part is seen on the right-hand end of Fig. 1.

3d. The mechanism for supporting and operating the expanding tool. This consists of a longitudinal cast-iron frame with central-feed screw for moving the carriage which supports the operating tool and a pedestal for supporting the shaft by which the operating tool is revolved. A view of this part of the machine is seen in Fig. 2, also taken from "The Iron Age" of February 16, 1905.

Power is derived from a 25-H.P. electric motor through a belt, or may be obtained from a countershaft if desired. By reversing the current the machine may be run in either direction.

The belt pulley is secured to a transverse shaft fitted with a worm, working in an oil bath, and gearing with a worm-wheel, which is secured to a sleeve working in a bearing of the pedestal, and transmits motion to the main shaft of the machine through feathers to allow the shaft to move longitudinally through the pedestal as it moves back and forth with the tool carriage, in which it is securely bedded by means of a fixed and a screwed collar. A projection of this shaft beyond the tool-carriage bearing is secured by two keys to the inside of a hollow cast-iron shank or sleeve in which the expander spindle surrounded by the flaring sleeve is carried.

To the free end of the shank is secured by means of two keys and two set screws the collar flange, to whose face the tool head is bolted, the tool head consisting of a hollow shell in which the expanding and flaring rollers are located. The tool heads are of different diameters for the various diameters of pipe to be operated upon, and each diameter of tool head requires its particular diameter of collar flange.

The tool head consists of a hollow cast-iron shell, conical for a short length at an angle of about 20 degrees at its fixed

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end for carrying the flaring rollers, and cylindrical through the remaining part for carrying the expanding rollers. The expanding rollers and flaring rollers, each five in number, are made of hard steel.

Inside of the cylindrical part of the tool head is located the expander cone of hard cast steel. This expander cone is secured to a spindle which extends back nearly the entire length of the hollow shank, the spindle being slotted near its inner end and fitted with a threaded key, which meshes through a slot in the shank with a finely-threaded feed nut surrounding the shank, the feed nut being held stationary by means of a friction band secured to the tool carriage, while the shank and tool head are revolving to feed in the expander cone and force out the expander rollers while expanding the pipe into its flange.

Back of the expander cone in the tool head is located the flaring cone, which is secured to a sleeve surrounding the expander spindle, and is fed in a similar way as the expander cone, by a second feed nut surrounding the shank, by means of a second friction band secured to the tool carriage.

These two feed nuts fit loosely on the shank and when not held by the friction bands revolve with it. They are longitudinally secured in position by two screwed collars which are secured to the shank each by two set screws, and by a fixed collar of the shank on the tool-head side. The collar between the two feed nuts rests on a roller bearing secured to the tool carriage to support the overhang of the revolving tool.

Gauges are fitted to the keys of the expander spindle and flaring sleeve by which the amount of work done by the expanding and flaring rollers can be ascertained.

A facing tool is secured near the free end of the cylindrical part of the tool head when it is desired to face off the end of the pipe in the flange.

The tool carrriage consists of a pedestal, which is movable along the frame of the machine, and can be moved in either direction by means of the central feed screw, either by a hand wheel on the side of the machine frame or by gearing from the

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end of the transverse pulley shaft. The tool carriage, however, is only moved to enter the tool head to its desired position in the pipe, and during the operations of expanding and flaring is thrown out of gear and remains stationary. It is fed by the hand wheel, however, while the end of the pipe is being faced off.

On February 14th a length of 20-inch welded-steel pipe, $\frac{3}{8}$ inch thick, together with its steel flange, had been secured in position in the machine just described, but without the hydraulic ram being in contact with the free end of the pipe. The tool head had also been secured to the machine.

After running the tool head into the pipe for position and starting the motor for turning the tool head, the feed for the expanding rollers was put on, and the pipe was expanded into its flange from 11·21 to 11·29 by the watch, or 8 minutes; the motor was then reversed and the expander rollers collapsed by 11·33 A. M., or 4 minutes more. The tool head was then run further into the pipe until the flaring rollers were in position, and the flaring of the pipe was completed by 11·39 A. M. The motor was again reversed and the flaring rollers collapsed by 11·42 A. M. The flaring rollers were then secured by screwed clamps and the expanding rollers by a belt by 11·44 A. M. A facing tool was then inserted into the tool head near its end and the end of the pipe was faced off even with the flange from 11·47 to 11·53 A. M., after which the tool head was withdrawn from the pipe and the machine stopped at 11·55 A. M.

The whole operation from the time the tool head had entered the pipe, of expanding, flaring and facing the pipe, from 11·21 to 11·55 A. M., consumed a period of 34 minutes. The surfaces operated upon remained cool to the touch. The expanded and flared surfaces of the pipe were perfectly smooth, the flared surface making an angle of about 20 degrees with the axis of the pipe. The pipe was left in the machine after the operation was completed.

For expanding and flaring pipes of smaller diameter the operating mechanism and jaws for holding the pipe flanges

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are mounted on a single cast-iron base, with a motor at the rear end, driving the machine by belt and cone pulleys or by means of countershafts over head provided with cone pulleys, as may be desired.

In general, these machines are made in three sizes—the smallest size for expanding pipe from 2 inches to 7½ inches diameter, the intermediate size for expanding pipe from 8 to 16 inches in diameter, and the largest size for expanding pipe up to 30 inches in diameter.

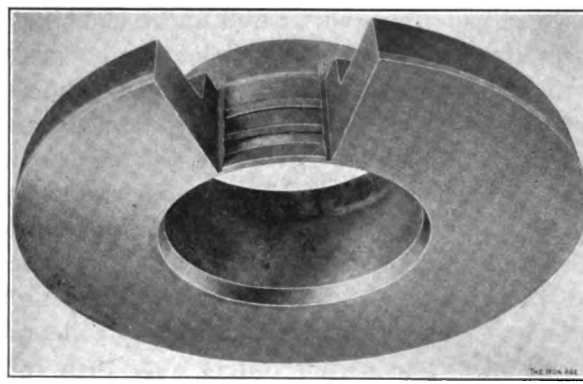

Fig. 3

A view of an expanded and flared pipe end is shown in Fig. 3.

The following tests were made on February 14th and 16th, 1905, to test the tightness of various pipes and flanges under hydraulic pressure, and then to determine the pressure required to force the flanges off their pipes.

To test the tightness under hydraulic pressure the pipes were blank flanged, as shown on the accompanying illustrations, and pressures were obtained by pumping with an hydraulic jack against the top crosshead of the pump frame, the pipe under test being connected with the hydraulic jack and a pressure gauge, as shown in the accompanying illustration, Fig. 4.

To make the second test, that of forcing the flange off its pipe, the blank flange was removed from one end of the pipe,

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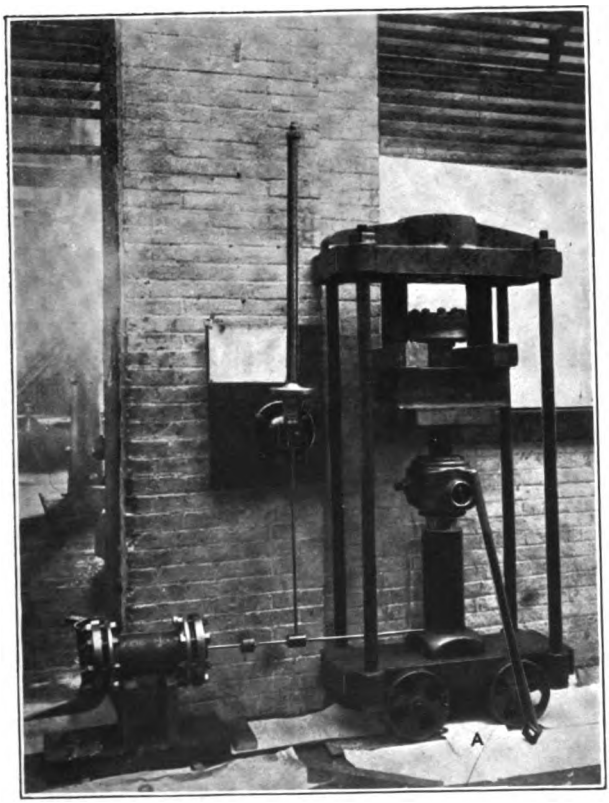

Fig. 4.

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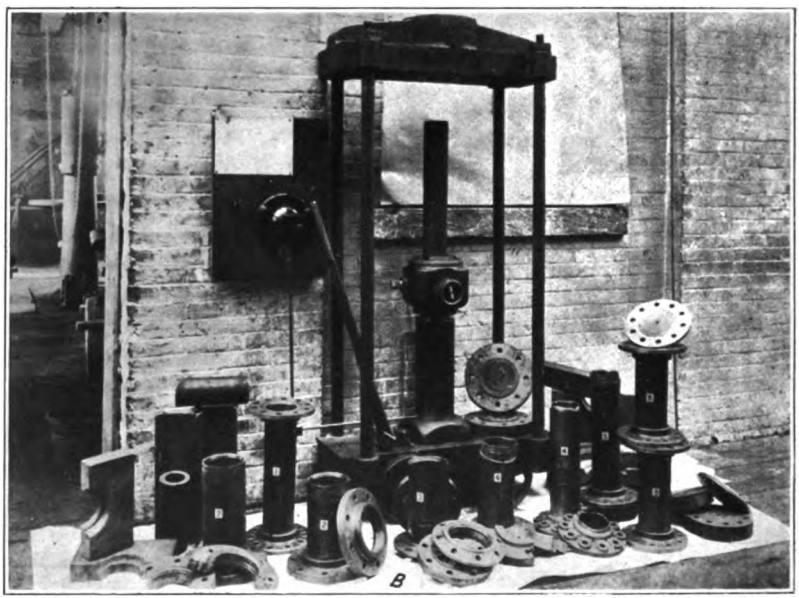

Fig. 5.

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and a metal bar, of somewhat greater length than the pipe, and of such diameter as to be loose in the pipe, was placed vertically on the head of the hydraulic jack, as shown on the accompanying illustration, Fig. 5, and was covered by the pipe under test, the upper end of the bar pressing against the blank flange at the upper end of the pipe, while the lower flange of the pipe was braced down from the upper crosshead of the pump frame, as shown in Fig. 4. Fig. 4 shows the hydraulic pump rigged for both of the above tests, but only one test was carried on at a time.

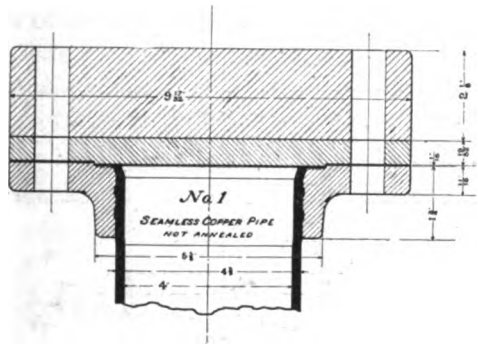

Fig. 6.

Pipe No. 1 (Fig. 6), consisted of a seamless copper pipe, 4 inches inside diameter, $\frac{3}{16}$ inch thick and about $18\frac{5}{8}$ inches long, expanded and flared without previous annealing into a forged-steel flange at each end, the flange with its collar at the bore being $1\frac{3}{4}$ inches thick and $\frac{11}{16}$ inch thick beyond the collar, and containing two rectangular grooves $\frac{3}{16}$ wide and $\frac{1}{32}$ inch deep. The flanges were counterbored at their face to an angle of 20 degrees for a length of about $\frac{1}{4}$ inch, and countersunk $\frac{1}{16}$ inch for a $5\frac{1}{2}$ inch diameter. The blank flanges were made in two parts, a forged-steel plate $\frac{21}{32}$ inch thick for a $5\frac{1}{2}$ inch diameter and $\frac{19}{32}$ inch thick outside of this diameter was backed by a cast-iron flange $2\frac{1}{8}$ inches thick; both were of an extreme diameter of $9\frac{17}{32}$ inches and $7\frac{1}{2}$ inches in diameter at bolt centers, and were bolted to the flange of the pipe by

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ten $\frac{3}{4}$-inch steel bolts, with a thin copper gasket placed in the counter-sunk part of the pipe flange.

On pumping up the cold-water pressure, one of the gaskets began leaking at a pressure of 3,200 pounds per square inch, and the pressure could not be raised above 3,600 pounds on account of this leak. After the pressure had been released, one of the pipe flanges was found loose on the pipe. This pipe was re-expanded in its flanges, and in making the second test for forcing the flanges off the pipe, the upper flange slipped over the pipe at a total pressure on the ram of 24.73 tons, produced by a pressure in the pump of 3,600 pounds per square inch. The lower flange had also just started to move.

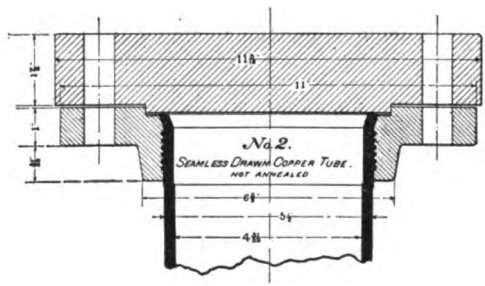

Fig. 7.

Pipe No. 2 (Fig. 7), consisted of a seamless-drawn 5-inch copper pipe, measuring $4\frac{31}{32}$ inches inside and $5\frac{1}{2}$ inches outside diameter and about $11\frac{3}{16}$ inches long, expanded and flared without previous annealing into a forged-steel flange at each end, the flange with its collar at the bore being $1\frac{31}{32}$ inches thick and 1 inch thick outside the collar, and containing six flattened V grooves, $\frac{1}{16}$ inch wide at the bottom, $\frac{1}{16}$ inch deep and $\frac{3}{16}$ inch pitch. The flanges were counterbored at the face to an angle of twenty degrees for a length of about $\frac{11}{32}$ inch, and countersunk $\frac{7}{32}$ inch for a $6\frac{1}{2}$-inch diameter; the face of the countersunk part containing four small circular V-shaped grooves for securing the gaskets. The blank flanges were of cast iron, were turned to fit into the countersunk part of the pipe flange, were $1\frac{7}{8}$ inches thick at

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the bolt holes, $8\frac{15}{16}$ inches in diameter at bolt centers, and $11\frac{5}{16}$ inches over all, and were secured by eight $\frac{3}{4}$-inch steel bolts with a thin copper gasket placed in the countersunk part of the pipe flange.

On pumping up the cold-water pressure, this pipe began leaking at a pressure of 1,800 pounds per square inch between the pipe and flange collar, and increased $\frac{1}{32}$ of an inch in length between center-punch marks, originally $10\frac{21}{32}$ inches apart, without increasing in diameter. After this pipe was re-rolled in its flanges it began leaking through the gasket at a pressure of 2,800 pounds per square inch, and through a porous spot in the blank flange, the expanded joints remaining tight. The diameter of the pipe had increased $\frac{1}{64}$ of an inch at this pressure, as had also the length between center-punch marks.

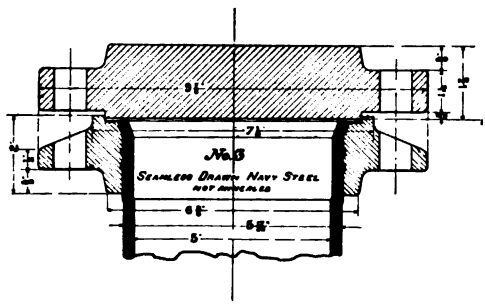

Fig. 8.

Without further re-rolling, this pipe was subjected to the second test for forcing off the flanges, and the upper flange slipped off the pipe at a total ram pressure of 15.188 tons, produced by a pressure in the pump of 2,200 pounds per square inch, the indications of the pipe showing that the pipe had been expanded only into the two lower grooves of the flange.

Pipe No. 3 (Fig. 8) consisted of a 5-inch seamless-drawn steel pipe, as supplied to the Navy, measuring $5\frac{9}{16}$ inches outside diameter, and $\frac{17}{64}$ of an inch thick, and about $12\frac{1}{8}$ inches long, expanded and flared, without previous annealing, into a forged-steel flange at each end, the flange with its collar at the bore

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being 2 inches thick, countersunk $\frac{1}{8}$ of an inch on its face to a $6\frac{1}{2}$-inch diameter, and cut away on its face $\frac{5}{16}$ of an inch beyond the $7\frac{1}{8}$-inch diameter, and tapering beyond this to a thickness of $\frac{1}{2}$ inch at its circumference of $9\frac{7}{8}$ inches diameter. The flanges were counterbored at the face to an angle of 20 degrees for a length of about $\frac{7}{16}$ of an inch, and the counterbore was countersunk for a length of about $\frac{5}{16}$ of an inch to form an extra groove for the flared portion of the pipe; the cylindrical bore of the flange was cut with two rectangular grooves, $\frac{5}{32}$ of an inch wide and about $\frac{3}{64}$ of an inch deep, the $\frac{3}{16}$ of an inch ridge between these grooves being rounded off to a radius of about $\frac{5}{32}$ of an inch. The countersunk face of the flange contained four small V grooves for holding the gasket. The blank flanges were of forged steel, were turned to fit into the countersunk face of the pipe flange, were $1\frac{7}{8}$ inches thick through $6\frac{3}{8}$ inches diameter and $1\frac{1}{16}$ inches thick at the outer portions about the bolt holes, and were seeured by eight $\frac{3}{4}$-inch steel bolts with a thin copper gasket placed in the countersunk part of the pipe flange.

On pumping up the cold-water pressure a leak occurred through one of the gaskets at 600 pounds per square inch, and increased to such an extent that it was impossible to raise the pressure above 3,800 pounds per square inch, the expanded joints showing no leak at this pressure. The pipe at this pressure had lengthened about $\frac{1}{64}$ of an inch.

After fitting the blank flanges with $\frac{1}{16}$-inch lead gaskets this pipe was again subjected to hydraulic pressure, and when a pressure of 4,100 pounds per square inch had been attained the gasket at the near end to the pump began leaking slightly. At a pressure of 4,800 pounds per square inch the flange at the far end of the pipe slipped at the top of the pipe, although there had been no leaks through the expanded joints up to this time. The pipe at this pressure had expanded $\frac{5}{32}$ of an inch in diameter near the middle of its length; the pipe had also increased about $\frac{21}{32}$ of an inch in a length of 10 $\frac{3}{16}$ inches.

After the pressure had been taken off completely, it was again pumped up, and the cock-billed flange began leaking

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through the expanded joint at a pressure of 950 pounds per square inch.

On applying the second test for forcing the flanges off the pipe the cock-billed flange slipped off the pipe at a total pressure of 34.36 tons on the ram produced by a pressure in the pump of 5,000 pounds per square inch, the pipe showing that the expanded metal had not been well forced into the flange grooves.

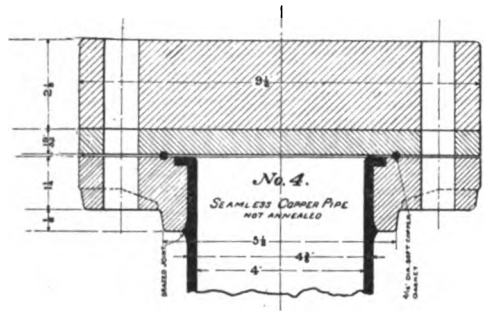

Fig. 9.

Pipe No. 4 (Fig. 9) consisted of a 4-inch seamless-drawn copper pipe 4 inches inside diameter, $\frac{3}{16}$ of an inch thick and about $18\frac{3}{8}$ inches long between flange faces. This pipe was previously annealed; was hammer pened and beaded by the usual old method, and was brazed to its composition flanges.

The flange with its collar at the bore was $1\frac{3}{4}$ inches thick, and tapered beyond the collar to a thickness of $\frac{3}{4}$ of an inch at the diameter of the bolt-hole centers, and continued at that thickness to the circumference, whose diameter was $9\frac{1}{2}$ inches. The back of the flange was bossed around the bolt holes to a thickness of $1\frac{1}{4}$ inches; the face was recessed to a depth of $\frac{3}{16}$ of an inch for a diameter of 5 inches to receive the beaded end of the pipe, and the face was further grooved at the $5\frac{1}{2}$-inch diameter to receive a $\frac{3}{16}$ of an inch soft copper wire for gasket.

The blank flanges were made in two parts, a forged-steel plate $\frac{19}{32}$ of an inch thick was backed by a cast-iron flange $2\frac{1}{8}$ inches thick at each end of the pipe, and were secured to the pipe flanges by ten $\frac{3}{4}$-inch steel bolts.

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On pumping up the cold-water pressure this pipe began increasing in length at a pressure of 900 pounds per square inch, and began leaking through the gasket in the far joint from the pump at a pressure of 1,800 pounds per square inch. The pressure was discontinued after 2,200 pounds per square inch had been reached, when the pipe had increased $\frac{1}{8}$ of an inch in an original length of $17\frac{3}{32}$ inches between center-punch marks on the flanges and $\frac{1}{4}$ of an inch in diameter.

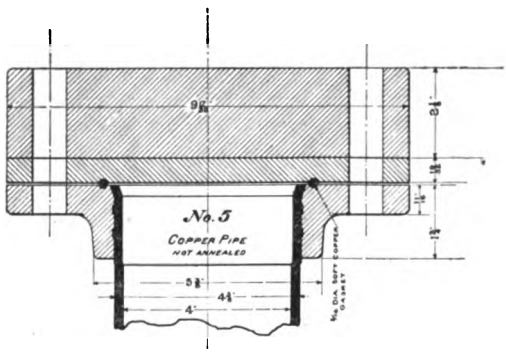

Fig. 10.

Subjected to the second test for stripping off the flanges, when a total pressure of 29.5 tons on the ram had been attained, produced by a pressure of 4,300 pounds per square inch in the pump, the pipe had stretched to such an extent that the center-punch marks had separated $1\frac{29}{32}$ inches, and the top flange was touching the crossbar of the pump frame. After readjusting the position of the pipe and again applying the pressure, the pipe broke about $\frac{1}{2}$ of an inch below the upper flange with a total pressure of 29.5 tons on the ram, the center-punched length of $17\frac{3}{32}$ inches had increased $2\frac{9}{32}$ inches, and the outer diameter of the pipe had been reduced to $4\frac{3}{8}$ inches. The brazing at the collars of the flanges was cracked lengthwise with the pipe in numerous places, and both pipe flanges were dished about $\frac{1}{4}$ of an inch, the circumference of the flanges being drawn $\frac{1}{2}$ of an inch further apart than the bores.

Pipe No. 5 (Fig. 10), consisted of a seamless-drawn 4-inch

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copper pipe, cut from the same length of pipe as pipe No. 1, 4 inches inside diameter, $\frac{3}{16}$ of an inch thick and about $18\frac{5}{8}$ inches long, expanded and flared without previous annealing into a forged-steel flange at each end, the flanges being of the same description and dimensions as those on pipe No. 1.

This pipe was not subjected to hydraulic pressure, but was only subjected to the second or pulling test to force the flanges off the pipe. In this test the lower flange slipped at a total ram pressure of 28.175 tons, produced by a pressure in the pump of 4,100 pounds per square inch, partially shearing off the extruded pipe metal in the flange grooves and slipping over it. The upper flange had slipped but slightly. This pipe stretched to such an extent that it closed in on the mandrel, and had to be forced off.

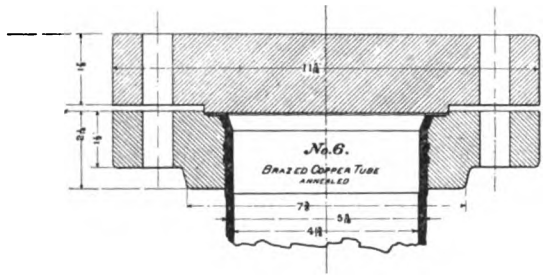

Fig. 11.

Pipe No. 6 (Fig. 11), consisted of a 5-inch brazed copper pipe, not uniformly round, measuring $4\frac{15}{16}$ inches inside diameter and $5\frac{5}{16}$ inches outside, and about $12\frac{5}{16}$ inches long, expanded and flared, after annealing, into a cast-iron flange at each end, the flange with its collar at the bore being $2\frac{1}{16}$ inches thick, and countersunk $\frac{1}{8}$ of an inch on its face to a $6\frac{1}{2}$-inch diameter. The flanges were counterbored at the face to an angle of 20 degrees for a length of about $\frac{7}{16}$ of an inch, and the counterbore was countersunk for a length of about $\frac{5}{16}$ of an inch to form an extra groove for the flared portion of the pipe. The cylindrical bore of the flange was cut with two rectangular grooves $\frac{5}{32}$ of an inch wide and about $\frac{3}{64}$ of

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an inch deep, the $\frac{3}{16}$ of an inch ridge between these grooves being rounded off to a radius of about $\frac{5}{32}$ of an inch. The countersunk face of the flange contained four small V grooves for holding the gasket. The expanded portion of this pipe, it will be seen, was in every respect the same as that of the 5-inch steel pipe No. 3. The blank flange was of cast iron, turned to fit into the countersunk face of the pipe flange, where it was $2\frac{3}{32}$ inches thick, and $1\frac{7}{8}$ inches thick beyond this boss, and was secured to the pipe flange by eight $\frac{3}{4}$-inch steel bolts.

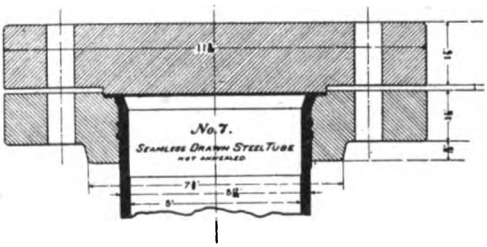

Fig. 12.

This pipe was not subjected to hydraulic pressure but was subjected only to the second or pulling test to force the flanges off the pipe. In this test when a total pressure of 34.36 tons had been attained on the ram, produced by a pressure of 5,000 pounds per square inch in the pump, the pipe broke throughout its circumference near the bottom of the flare in the lower flange, except through a circumferential length of about $2\frac{1}{4}$ inches where the break extended about $1\frac{1}{2}$ inches further from the end. This lower flange was later broken by a sledge. An examination of the pipe showed that the grooves of the flange had been well filled by the expanded portion of the pipe.

Pipe No. 7 (Fig. 12), consisted of a 5-inch seamless-drawn steel pipe, of the same character as pipe No. 3, and measured 5 inches inside diameter, $5\frac{17}{32}$ inches outside diameter, and about $12\frac{5}{16}$ inches long. It was expanded and flared without previous annealing into a cast-iron flange at each end, the flanges being in every respect the same as those of Pipe No. 6, excepting that due to slight difference of the outside diame-

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ter of the pipes. The blank flanges were also of the same material and dimensions as those of Pipe No. 6, and they were similarly secured by eight $\frac{3}{4}$-inch steel bolts.

This pipe was subjected only to the second or pulling test to force the flanges off the pipe. In this test when a total pressure of 32.98 tons had been attained on the ram, produced by a pressure of 4,800 pounds per square inch in the pump, the lower flange slipped, slightly shearing off the extruded collars of the expanded pipe, the metal of which had not been well forced into the flange grooves during the expansion. The upper flange was not started. The flanges of this pipe were then broken with a sledge.

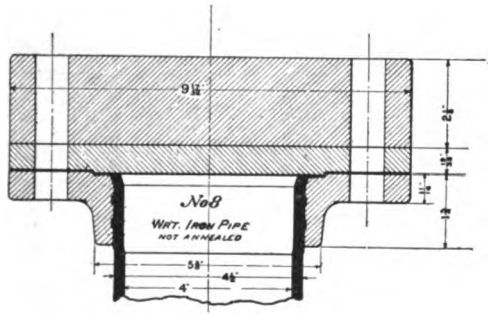

Fig. 13.

Pipe No. 8 (Fig. 13), consisted of a 4-inch wrought-iron pipe, measuring 4 inches inside diameter, $4\frac{1}{2}$ inches outside diameter and about 13 inches in length, expanded and flared without previous annealing into a forged-steel flange at each end, the flange with its collar at the bore being $1\frac{3}{4}$ inches thick and $\frac{11}{16}$ of an inch beyond the collar. The face of the flanges were countersunk $\frac{1}{16}$ of an inch to a $5\frac{1}{2}$-inch diameter and contained four small V grooves for holding the gaskets. The flanges were counterbored at the face to an angle of 20 degrees for a length of about $\frac{1}{4}$ of an inch; the cylindrical part of the bore was cut with two rectangular grooves $\frac{3}{16}$ of an inch wide and $\frac{1}{32}$ of an inch deep with a ridge $\frac{3}{8}$ of an inch wide between them. The blank flanges were made in two parts, a forged-steel plate $\frac{11}{16}$ of an inch thick to fit into the countersunk part

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of the pipe flange and $\frac{19}{32}$ of an inch thick beyond that was backed by a cast-iron flange $2\frac{1}{8}$ inches thick and secured to the pipe flange by ten $\frac{3}{4}$-inch steel bolts.

This pipe was submitted to the hydraulic test only, and is said to have been re-rolled after a previous pressure of 3,600 pounds per square inch. At the present test it began to leak under the flange away from the pump when the pressure had reached 2,000 pounds per square inch, and the flange slipped $\frac{1}{8}$ of an inch toward the end of the pipe.

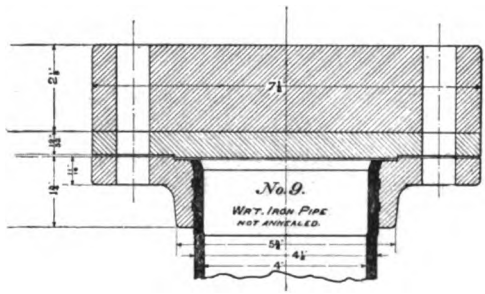

Fig. 14.

It will be seen that pipes Nos. 1, 2, 3, 4 and 8 were subjected to the hydraulic test to determine their capacity to withstand leaks from within, and that pipes 1, 2, 3, 4, 5, 6 and 7 were subjected to the ram test for stripping the flanges off the pipes. Thus, while pipes 1, 2, 3 and 4 were subjected to both tests, pipes 5, 6 and 7 were subjected to the stripping test only, and pipe No. 8 was subjected to the hydraulic test only. Pipes Nos. 1, 2, 4, 5 and 6 were of copper, while pipes 3 and 7 were of steel, and pipe No. 8 of wrought iron. Pipes 1, 4, 5 and 8 were 4-inch pipes, and pipes 2, 3, 6 and 7 were 5-inch pipes. Pipes 1, 2, 3, 4, 5 and 7 were seamless-drawn, pipe No. 6 a brazed copper, and pipe No. 8 a lap-welded iron pipe. Pipe No. 4 was hammer pened and beaded by the usual old method, and brazed to its composition flanges, while all the other pipes were expanded and flared by machine. The flanges were of various materials; those of pipes 1, 2, 3, 5 and 8 were of forged steel; those of No. 4 composition, and those of 6 and 7 of cast

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iron. With the exception of the flanges of No. 4 pipe all were grooved in the bore and flared at the face. Those of pipes Nos. 1, 3, 5, 6, 7 and 8 were each fitted with two rectangular grooves, the ridge between these grooves being of rectangular section on Nos. 1, 5 and 8, while on Nos. 3, 6 and 7 this ridge was rounded off to facilitate the flow of the extruded pipe metal into the grooves. The flanges of No. 2 pipe were fitted with six flattened V grooves.

The following is a synopsis of the pipe tests.

HYDRAULIC TEST.

No. 1 Pipe.—Lead gasket began leaking at 3,200 pounds and pressure could not be raised above 3,600 pounds, and one flange became loose.

No. 2 Pipe.—Began leaking between pipe and flange at 1,800 pounds pressure per square inch, and after re-expanding pipe in flange it began leaking through gasket at 2,800 pounds.

No. 3 Pipe.—Began leaking through the gasket at 600 pounds and pressure could not be raised above 3,800, there being no leak between pipe and flange. After fitting new lead gasket, one gasket began leaking at 4,100 pounds pressure per square inch, and one flange slipped at 4,800 pounds pressure per square inch.

No. 4 Pipe.—Began leaking through one gasket at 1,800 pounds per square inch, and pressure discontinued at 2,200 pounds per square inch.

No. 5 Pipe.—No test.

No. 6 Pipe.—No test.

No. 7 Pipe.—No test.

No. 8 Pipe.—This pipe is said to have been re-rolled after having been subjected to a previous test of 3,600 pounds per square inch. At present test began to leak between pipe and one flange at 2,200 pounds per square inch and flange began slipping.

STRIPPING TEST.

No. 1.—After expanding pipe in flange, upper flange slipped at 24.73 tons ram pressure.

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No. 2.—Upper flange slipped off pipe at 15.188 tons ram pressure. Pipe metal did not extrude into all the grooves, only the two lower grooves being full.

No. 3.—Cock-billed flange was slipped off pipe at ram pressure of 34.36 tons. Pipe metal had not been well forced into flange grooves.

No. 4.—At 29.5 tons ram pressure pipe had stretched enough to touch crosshead on top of frame, and after re-adjusting its position, when a ram pressure of 29.5 tons was again attained, the pipe broke about one-half inch below top flange.

No. 5.—Lower flange slipped at ram pressure of 28.175, tons, partially shearing off extruded pipe metal and partially slipping over it.

No. 6.—Pipe broke throughout circumference near the inner end of the flare in lower flange at a total ram pressure of 34.36 tons. Grooves had been well filled with extruded metal.

No. 7.—Lower flange slipped slightly at a total ram pressure of 32.98 tons, shearing off the extruded collars, which had not well filled the grooves.

No. 8.—No test.

The object of the pipe tests was to show the superior strength or holding power of the flanges of the machine-expanded pipe over that of the hammer-pened and brazed one.

It will be seen from the hydraulic tests that of the machine-expanded pipes No. 2 began leaking between flange and pipe at a pressure of 1,800 pounds per square inch, and after re-expanding was tight between pipe and flange when the pressure was discontinued at 2,800 pounds on account of a leaky gasket and porous blank flange; that a flange on No. 1 pipe became loose at a pressure of 3,600 pounds, of No. 3 at 4,800 pounds and of No. 8 at 2,200 pounds, while the hammer-pened and brazed pipe No. 4 was tight between pipe and flange when the pressure was discontinued at 2,200 pounds on account of a badly leaking gasket.

In the ram or stripping tests one flange became loose by slipping on each of the machine-expanded pipes Nos. 1, 2, 3, 5 and 7, at total pressures varying from 15.188 tons in No. 2

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to 34.36 tons in No. 3, the stripping pressure being 24.73 tons for No. 1, 28.75 tons for No. 5, and 32.98 tons for No. 7, while the machine-expanded pipe No. 6 broke throughout its circumference near the inner end of the flare in one of its flanges at a total pressure of 34.36 tons, and the hammer-pened and brazed pipe No. 4 broke throughout its circumference about ½ inch below its top flange at a total pressure of 29.5 tons.

The greater elongation of No. 4 pipe, while undergoing the stripping test, was undoubtedly due to its having been annealed before it was hammered into its flanges. The elongation of No. 6 pipe, similarly annealed before expanding, was not ascertained. None of the other pipes were annealed before expanding.

Of the machine-expanded pipes the pipe metal was not well extruded into the flange grooves of pipes Nos. 2, 3 and 7, and of these, No. 2, a copper pipe, was the only pipe whose flanges were fitted with the flattened V grooves, and pipes Nos. 3 and 7 were of steel, which metal flows less readily than copper.

It can be readily seen, however, that the method of fitting pipes into their flanges by the expanding machine is much superior to the old method of hammer pening, the time necessary to perform the same operation is by it very much reduced, it causes much less strain on the metal of the pipe, and makes the work much more reliable. Of course, care must be taken that the expanding tool is centrally located within the pipe in expanding, and that the expanding is not carried so far as reduce the thickness of the metal of the pipe.

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SELECTION AND USE OF CERTAIN APPARATUS AND MATERIAL.

By Ensign H. T. Winston, U. S. N., Member.

Compiled from the files of Bureau of Steam Engineering and other sources.

It is intended in this article to show how the best results may be obtained on board U. S. naval vessels with certain material and apparatus; the experience (or lack of experience) of those who care for and operate the machinery, and the general service conditions as they are at present being considered, and the special exceptional cases where some particular arrangement might be satisfactory or perhaps slightly superior being disregarded.

Where there is considerable difference of opinion on any subject, the majority opinion is given.

PIPE AND BONNET JOINT PACKINGS.

High-Pressure Pipe Joints.—For steam pressures over 200 pounds, corrugated, annealed, soft-copper gaskets give best results. The flanges should be male and female, and at least as heavy as the Bureau of Steam Engineering standard (standard sheets Nos. 6, 7 and 9), accurately fitted and grooved; the gaskets and flanges being lightly coated with red lead, care being taken that the quantity used is not sufficient to fill the corrugations and thus destroy their purpose.

For pressures below 200 pounds, less difficulty is experienced in making a tight joint. While the joint mentioned above has certain advantages on account of its lasting qualities, yet on the whole the sheet packings mentioned in the following paragraph have been found more satisfactory.

Whenever the high grade of workmanship necessary for making a copper gasket joint cannot be obtained, where the

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flanges are light or old and poorly fitted, where there is considerable vibration, as on torpedo boats, or other causes for not using copper gaskets exist, a waterproof, braided, asbestos sheet packing with wire insertion should be used.

Such packings as Gladiator, Kearsarge and Vulcabeston have given best results.

Water Pockets.—Some trouble has been experienced where steam condenses and water collects at the joint; in such cases a rubber-composition, wire-insertion, sheet gasket may have to be used. Rainbow and brown Garlock have given satisfaction, while in some instances Melville's white oxidized packing has been highly recommended.

A red-lead putty joint with copper or brass wire gauze gives fair satisfaction for pressures below 160 pounds.

Small Branch Pipes.—As these pipes are subject to vibration and jar, flanges as a rule not accurately fitted, bolts not so well spaced and joints frequently broken, copper gaskets should not be used. Asbestos or rubber composition, with wire insertion, will be found more satisfactory.

High-Pressure Bonnet Joints.—For small, well-fitted bonnets plain copper gaskets give satisfaction.

In general, bonnet joints, other than those fitted with copper gaskets, are found most satisfactory when fitted with asbestos sheet material, such as Gladiator and Kearsarge.

Wet joints, however, as those on separators and steam traps, should be made with a high-pressure, wire-insertion, rubber-composition packing, similar to Rainbow, Garlock or Usudurian.

McKim's gasket, King's packing, Klingeret, Melville's white oxidized, Libby's IXL and Common Sense packings have been recommended by some engineers for various bonnet joints.

Low-Pressure Steam and Exhaust Lines (pressures below 50 pounds).—Owing to their durability, copper gaskets should be used where flanges are heavy and well fitted, particularly where the pipes are large and joints inconveniently located for the renewal of gaskets.

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A rubber-composition, wire-insertion packing, similar to Rainbow, Garlock or Usudurian, is the best well-known material for joints other than those mentioned above.

A waterproof asbestos-composition, wire-insertion packing will give good results where the joint is fairly free from water.

High-Pressure Fresh-Water Piping.—(Pressures over 175 pounds.)

Bonnets.—For bonnets, joints of feed-water heaters, feed pumps, grease extractors, etc., the following give best results, preference being given in the order shown:

(a) Sheet-rubber composition with wire or cloth insertion, similar to Rainbow, Garlock or IXL.

(b) Soft, corrugated copper gaskets or rings of soft copper wire, where joints are well fitted and not subject to excessive vibration.

(c) Copper or brass wire gauze and red lead, great care being taken in making the joint and in avoiding the presence of lumps in the red-lead putty.

Feed Lines.—For boiler-feed lines preference is given in the order shown below:

(a) Corrugated soft copper gaskets;

(b) Cloth- or wire insertion rubber sheet, such as Rainbow, Garlock or Usudurian;

(c) Waterproof sheet asbestos composition with wire insertion, similar to Gladiator or Kings;

(d) Red lead with copper or brass wire gauze, when carefully fitted.

Melville's white oxidized packing has been highly recommended in this connection by several engineers.

Low-Pressure Fresh-Water Joints—(below 150 pounds).—Any good quality of sheet gum with cloth insertion will give satisfaction. For pressures below 50 pounds the cloth insertion is not necessary.

Low-Pressure Salt-Water Joints.—The packing mentioned in the preceeding paragraph is satisfactory. In renewing these gaskets on board ship, however, it would probably be found desirable to use canvas soaked in linseed oil and

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painted with a thick coat of red lead, on account of the cheapness of material, wide-spread use of same for other purposes, and slight deterioration while in store as compared with gum sheet.

NOTES ON RENEWING GASKETS AND FITTING JOINTS WITH SHEET PACKING.

In addition to the preceding, the following suggestions might be added:

When a corrugated copper gasket has been used once it will generally have to be re-annealed and the corrugations reformed in the former before using again, especially in the case of a joint which is likely to leak.

Sheet packing should be chosen as thin as the smoothness of flange surfaces and accuracy of fitting will permit, to prevent likelihood of gasket blowing out; but in case the joint surfaces are not parallel or not properly finished, a very thin gasket would not make a tight joint, and cannot be used.

Ordinary asbestos sheet should never be used for a joint where there is apt to be an accumulation of water or wet steam.

Care must be exercised to see that the gasket is cut properly, no openings being blanked by failure to cut holes in gasket, bolt holes cut so as to give a snug fit around bolts, and that the part which is covered by joint surface is free from cracks or cuts.

Use of Graphite, etc., on Gaskets.—In order that a gasket may be used over again after a joint is broken, and to facilitate the smooth breaking of the joint, graphite and tallow or graphite and cylinder oil should be spread on one or both sides.

Prepared graphite grease is perhaps better for this purpose than an improvised mixture. It would be well to note in this connection that tallow is objected to by some on the ground that the contained acid will attack the gasket material.

In the case of cylinder heads and other bonnet joints which

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are frequently broken, red lead should be used on one side of the gasket, as it will then adhere to one of the flanges.

Red lead may be used on both sides of gaskets for cold-water piping, but should not be used for hot water, owing to the difficulty in breaking the joint.

Neither sheet lead nor red lead should be used in lines which carry drinking water.

Wire gauze made from material which will corrode should not be used as an insertion for packing nor with red lead in making a joint.

Preservation of Packings in Store Rooms.—Copper gaskets require little care to prevent deterioration of the material, can be annealed if they harden, and may be corrugated by means of a former which is carried on board ship.

Copper wire, wire gauze, canvas, red lead and linseed oil also suffer little from deterioration.

Asbestos sheet material keeps better than rubber compositions. It should be kept in a cool, dry place to prevent mildewing.

Rubber compositions soon become hard and brittle when stored in a hot, dry place—the usual condition of a ship's storeroom. They should be kept in a cool place, and the stock on hand replenished frequently instead of carrying a large reserve supply.

SELECTION OF PACKINGS FOR NAVAL USE.

It is manifestly an advantage to have as few *different types of packings* as possible on each individual ship and also for the Navy as a whole.

An ideal arrangement would be to make yearly contracts for certain brands which are known to be satisfactory, and to use no other during that year except for experimental purposes, no change to be made until some new packing proves superior to those in use, when that brand should be called for in the next yearly contract. The contractors should be required to furnish the packings to some *central distributing station* (New York Navy Yard, for example), at intervals of three or

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four months, or as required by that station; suitable storerooms being provided for preserving packings in good condition, with good facilities for quickly issuing same on approved requisitions. All approved requisitions from ships and stations to be sent to this central station, which would issue fresh, well-preserved packings at frequent intervals for use throughout the Navy. The original outlay would be more than offset by the results obtained and decreased expenditures in the future.

It is believed that not more than four or five varieties need be kept on hand; these to be the ones which have been found most satisfactory in actual service.

It appears that certain packings, like other patented manufactured materials, are likely to be of a poorer quality after the necessary reputation has been made.

FILTERING MATERIAL FOR FILTER TANKS.

Coke put up in bags of coarse toweling, burlap or blanket material gives good results, though the coke disintegrates in time and gets into the feed pumps. With good, strong bags and frequent renewal, this trouble is largely obviated.

Excelsior sewed up in bags of material similar to the above may also be used.

Loofa, the vegetable filtering material, is in high favor with several engineers. It may be put in the filter tank loose or in bags, to suit special conditions.

Marine sponges should never be used as they rapidly disintegrate, are efficient for only a very short time, and are very expensive.

Considerable improvement can undoubtedly be made in the filter tanks at present in use with regard to efficiency and facility for overhaul, cleaning and renewal of filtering material.

Special care should be exercised that the filter chambers of tanks are easily accessible, and, when possible, the cover should be fitted with hinged bolts to facilitate removal.

It is important that filter tanks be cleansed frequently and the filtering material renewed when necessary; once a month

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will probably be found often enough under ordinary service conditions.

Drain pipes of every description should discharge into the filter tank, in order that all water taken from the feed tanks by the pumps will have been filtered.

Grease Extractors.—The following is from a circular recently issued by the Bureau of Steam Engineering:

Grease extractors, in addition to the separate filter tank, are in general to be fitted on all vessels building and on vessels undergoing general overhauling, where circumstances make it desirable and where suitable space and arrangement for their installation may be obtained.

Grease extractors will be located, in general, on the discharge side of the main feed pump, between pump and feed heater, or, in vessels where hot-well pumps are fitted, on the discharge side of hot-well pumps. The location must be such that all parts are accessible for overhauling and cleaning. Especial attention must be given to the details, so that bonnets and filter cartridges can be readily removed and the cartridge eased off its seat by mechanical means.

Grease extractors will be of the cartridge type, similar in principle to those shown on Bureau Steam Engineering plan No. 8, S, 1, on opposite page. They are to be fitted with by-pass valves.

The cartridge should be lined with coarse, heavy toweling or linen filtering cloth, and suitable spare filter jackets will be supplied. The surface of filtering jackets should be as follows: Ratio of area of filter jacket to area of feed pipe, not less than 30. The perforations of filter basket must be at least 50 per cent. of its total area.

Grease extractors will have a drain cock and a ½-inch steam connection for boiling out, a plug for introducing soda, and an air cock.

Grease extractors should have their filtering material renewed at least once every two weeks, if found necessary, and more frequently if a considerable quantity of grease and oil collects. They should be drained of oil, by-passed, and blown

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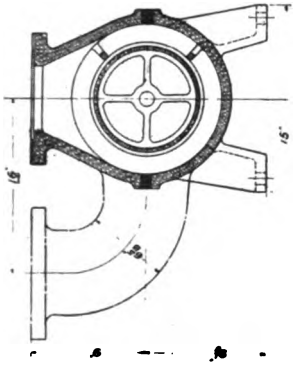

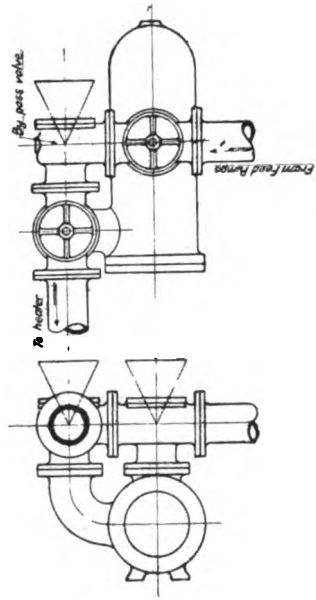

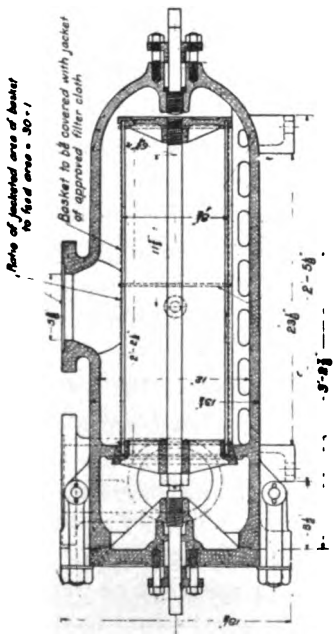

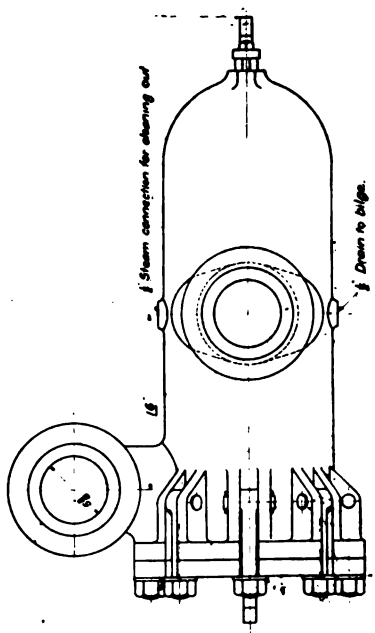

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out by steam daily while steaming and twice weekly when in port, if necessary. Except when under repairs, they will be kept in continual use.

STEAM TRAPS.

TYPES OF TRAPS.

The *open-bucket float type*, though bulky and somewhat affected by the ship's motion in a sea way, has given the most general satisfaction. It will take care of a sudden gush of water and requires no adjustment for varying pressures and temperatures.

The *hollow-ball float type* has given some trouble on account of the ball corroding through, getting punctured or becoming detached. This type should give satisfaction if proper provision has been made to insure watertightness of the float and if the float is securely attached to its lever.

The *expansion type* of trap is superior to the float type in that it is not bulky, has few parts and lasts well. It does not work well where there are wide variations in the temperature or amount of water to be handled. Under such conditions trouble has been experienced with the frequent adjustments, irregular working of trap and leakage of steam.

The following traps are largely used in the Navy: (float type) Dinkel, Nason, McKellar and Kieley; (expansion type) Geipel.

Requirements of a Good Trap.—The best trap is one that will discharge the water which collects without wasting steam and at the same time requires little attention. Other qualities to be considered are durability, simplicity, facility of overhauling, weight, size and cost.

It is better to have the drain valve of a trap at the top of the chamber rather than at the bottom, where sediment collects. It is also an advantage to have valves outside the float chamber, where they may be cleaned and overhauled without breaking a large joint.

The bonnet of a float trap should be secured with studs, or if through bolts are used, the heads should be so fitted as to

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prevent turning. The facility for overhauling might be still further increased by the use of hinged bolts.

The piping for all traps should be fitted with union couplings to facilitate the removal of trap for examination or repair. In the case of an expansion trap, the discharge should be fitted with a by-pass to the bilge by means of which it could be determined whether the trap was working properly.

Traps for Special Uses.—Where the quantity of water is subject to sudden variations, as in steam-pipe and separator drains, the float type should be used. Where there is a slow, regular and continuous condensation at a practically constant temperature, the expansion type is satisfactory and often desirable.

For *cylinder jackets* of main engines an arrangement similar to that on vessels of recent designs seems best; *i. e.*, steam goes from the H.P. jacket to the I.P. jacket through a reducing valve, from the I.P. jacket to the L.P. jacket through another reducing valve, and then drains through the L.P. jacket trap; one trap taking care of the water for the whole system.

The following easily improvised substitute for a jacket trap has been used with success: A small cast cylinder, fitted with a drain valve and gauge glass, is placed in the jacket drain-pipe line. When in operation the drain valve is opened just enough to keep the water in gauge glass at a fixed height, the whole arrangement being near the starting platform, and readily seen.

For *steam radiators* the bucket type generally gives best results. Expansion traps when used should be placed close to the radiators, and each trap should not drain more than two or three coils; they have been found to give considerable trouble when required to drain more than one heating circuit.

For *evaporators and feed heaters* bucket traps have been extensively used. It is probable that an expansion trap, carefully installed, would work efficiently on evaporator coils.

Drains for feed-water heaters when discharging to condensers do not require traps, but if the heater drains discharge to feed tanks traps may be necessary.

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For *steam separators and steam lines* the bucket type must be used.

LOCATION OF TRAPS.

A trap must be located below the lowest part of the vessel or pipe to be drained, and, if practicable, above the level of the tank into which it discharges, in order to prevent water backing up when the trap is not in operation. This applies particularly in the case of the expansion trap, which has given trouble in discharging against a head of water. Under certain conditions it may be necessary to fit a non-return valve in the discharge line.

All traps, and particularly those which will need frequent overhauling, should be so placed as to be easily overhauled and examined. If a trap cannot be installed where it is fairly accessible for overhauling it should be omitted and a drain valve fitted instead.

OPERATION AND OVERHAULING.

Many traps are of no value on account of lack of care and attention.

A float trap can be heard "dumping" when in order, and may work satisfactorily for months without overhauling. Whenever the sound of dumping is not heard at intervals the trap should be examined at once. Less trouble is experienced with a large trap than a small one, as in the latter case valves are more easily clogged, thus causing a failure to discharge, or if not entirely stopped will permit a leakage of steam. This is paritcularly the case where there is likely to be sediment in the water.

While experience is necessary to judge in each particular case the period a trap may be left without examination, it may be said in general that a large trap should never go more than three months nor a small trap more than two months without examination and cleaning.

By opening the expansion-trap discharge by-pass to bilge it may be seen whether the trap discharges water properly, and if it leaks steam. The trap should be tested frequently when in use and adjusted or cleaned out if necessary.

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THE FUTURE OF MARINE GAS ENGINES.

By Peter Eyermann, Associate.

In presenting this paper to a society interested in marine engines it is hoped that it may have the effect of attracting more attention to the gas engine, especially in so far as it relates to its possible substitution for the old and reliable reversible steam engine on board ship.

During the past century the progress of science and industry was marked by the ingenious development of the steam engine and boiler, and in the present one by the remarkable improvements already made in the gas engine.

As in a large measure all modern transportation facilities were developed along with the steam engine, may we not look for a similar advance as the gas engine and producer are perfected?

It will be noted that the principle is not a new one, it having been first brought forth about the year 1860, based upon the experiments of the French engineer Lenoir. His engine was, however, not sufficiently economical for commercial purposes.

The next step of importance in this connection was made in the Otto engine, brought out in Germany in 1882. The name of Lenoir will always be associated with the so-called "two-cycle" principle and double-acting engine, while that of Otto will be remembered in connection with the "four-cycle" principle and single-acting type. Both systems are so well known that any description of them would seem unnecessary. The main features will, however, be set forth in a few words.

Figure 1 (not in proper proportion) is the diagram for a four-cycle engine. Line I represents the suction stroke of the piston, during the whole of which the explosive mixture is being drawn into the cylinder; II is the second or compression

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stroke. III indicates the explosion which takes place at the beginning of the expansion stroke, IIIa, which is similar to the corresponding stroke in a steam engine. Before the end of this stroke is reached the cylinder begins to open to exhaust, as indicated at IIIb, and the whole of the following one, IV, is spent in expelling the products of combustion.

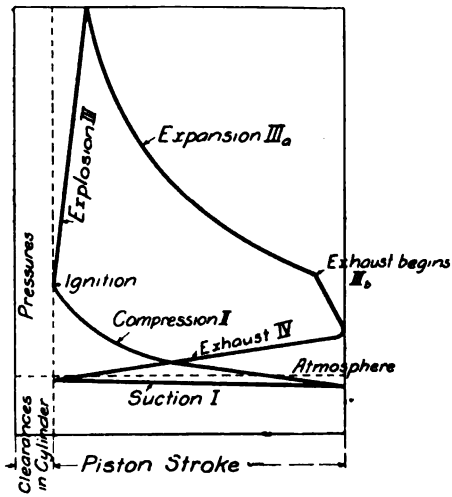

Fig. 1.

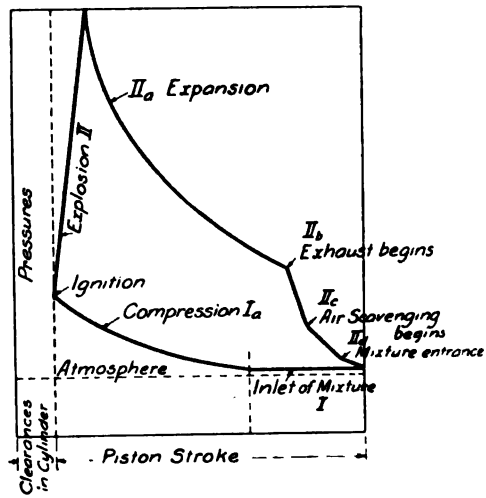

Fig. 2.

The various machines to be found on the market differ very little in so far as the principle of burning the gases is concerned, and little difference will be noted in the various diagrams. The main point of difference is in the crank angles, but a variation of ten or fifteen degrees one way or the other at the dead point of the piston will not be very noticeable on the diagram.

Fig. 2 represents the diagram for a two-cycle engine, and shows very clearly the difference between the two strokes of this type as compared with the four strokes shown on figure 1. During the first stroke the explosive mixture is forced into the cylinder (I) and compressed (Ia), while during the next, five operations are performed: these are: the explosion (II), followed by the expansion (IIa), the exhaust (IIb), the

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scavenging of the cylinder (IIc), and the admission of the fresh charge (IId).

It will be noted that in the case of figure 1 there are three lines below the atmosphere, from which it will be seen that the four-cycle engine is capable of sucking the explosive mixture into the cylinder, and may thus be made to produce gas from its own power. In figure 2 there are no lines below the atmosphere, from which it follows that some auxiliary power must be utilized for supplying gas and forcing it into the cylinder. These facts might show why, under certain circumstances, one type is to be preferred to the other. In connection with this much discussed question I would suggest that, as a general rule under ordinary circumstances, the two-cycle type be adopted where the gas is produced under pressure, and naturally the four-cycle type where the gas is to be produced by suction.

The use of explosion engines in small boats is so well known that little need be said concerning them. Up to the present time the only successful fuels for these engines have been of a liquid character, such as gasoline, kerosene and alcohol, though tests have been made with illuminating and other gases stored in tanks under a high pressure.

The use of gas producers for boats has, I believe, received attention since the earliest days of gas-engine building, but there have not been any very important developments along this line, and large gas engines for this purpose have not as yet been built. It appears that one of the principal reasons why the gas engine has not come into more general use is the uncertainty as to the question of reversibility. It has been held that engines of this type having a capacity of more than 100 H.P. are impracticable, it having been shown that, as compared with the ordinary steam engine, they are subject to much greater vibration, have much more weight in themselves, besides the heavy flywheel and gearing, and that the valve and igniter motions are of a complicated character. Owing to a lack of positive connection between the main engine and propeller shafting, it has been considered doubtful whether large

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gas engines would ever be available for use on board ships of the Navy.

The development of the large engine using a very poor lean gas has been such that I believe the builders of marine engines will be forced to take up this type, as was the case in stationary practice.

So far as I can recall there is not a large company in Europe, having a reputation for high-grade steam engines, which has not been forced to put on the market some type of internal-combustion engine.

So much has been written of late concerning the rapid development of this type of motor that I will not dwell upon that point, but would mention the fact that the improvement in the large types has been followed by a corresponding development of the producer, just as in the case of the steam engine where the increasing efficiency demanded better boiler and condensing plants. The best type of gas engine would have no sale if it were not possible to supply a cheap gas of good quality for its operation. The construction of large engines operating with blast-furnace gas having a calorific value of not more than 90 to 100 B.T.U.s has led to the development of another type to use producer gas whose value in B.T.U.s varies from about 125 to 170.

Fig. 3.

Kind of Gas.	H.	CH_4.	CO.	CO_2.	N.	Combustible parts.	Air required.	B. T. U.
Natural	2.	93.8	0.6	0.6	3.	96.4	8—9	1,100
Illuminating	46.	45.	7.5	1.	0.5	98.5	6—7	700
Coke oven	60.	22.	10.6	3.	4.4	92.5	4—5	450
Water	50.	6.	35.5	3.5	5.	91.5	3—4	350
Producer, bituminous coal	18.	3.	20.	4.	55.	41.	2.	160
anthracite coal	12.	1.	26.	5.	56.	39.	1.5	140
coke	3.	0.6	25.	6.	65.4	28.6	1.	120
Blast-furnace	2.	1.	24.	7.	63.	27.	0.9	100

Fig. 3 gives the composition of the above as compared with the rich natural and illuminating gases. This table substan-

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tiates the theory and is in keeping with the result of actual practice, where it has been found that the amount of air required for combustion increases with the quantity of combustible gases, and especially with the amount of heavy hydrocarbons in the mixture, and explains why poor gases have been found to give very nearly as much horsepower as stronger ones in cylinders of the same size.

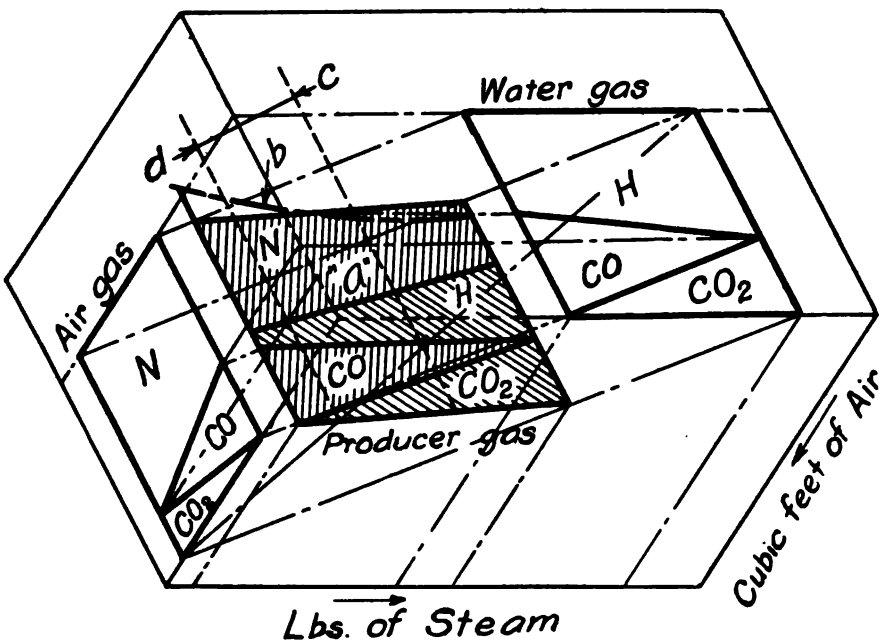

Fig. 4.

It has been demonstrated by special tests as well as in actual practice that, in the case of water-cooled engines, the more nearly instantaneous the combustion the better will be the efficiency obtained. Hydrogen being a highly-inflammable and quick-burning gas, is of assistance in this connection. The presence of a certain percentage of free hydrogen is also desirable to insure the proper action of the igniting spark.

Mr. K. Kutzbach, a German engineer, has recently published a very interesting paper in which he shows the influence

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of steam or vaporized water in gas producers, as follows: A diagram, the one at the left in figure 4 illustrates the gasifying of carbon with air. The one at the right is a similar diagram, in which steam is used. By connecting the two we obtain a third, a, showing the composition of ordinary producer gas It will be seen that producer gas consists, essentially, of nitrogen, hydrogen, carbon monoxide, and carbonic acid, and that the percentage of each is dependent upon the others.

The more hydrogen in the gas, the more carbon will be required for its manufacture and the more expensive it will be; but it will be seen from line b that the number of heat units is increased with the increase of the CO, and hence a better quality of gas is obtained. As stated above, the inflammability is increased by the addition of hydrogen; hence, for good gas-engine practice, the object should be to obtain a mixture with just enough hydrogen to insure ignition. With the producer a gas is generally obtained whose composition is represented by that portion of the diagram lying between the lines c and d. Blast-furnace gas is nearly an ideal one for gas engines, its composition being principally CO and nitrogen, while water gas is composed principally of CO and hydrogen.

Touching now upon the question of economy, which together with that of a reduction of the number of parts is of most importance on board a man-of-war, it will be noted that in the case of engines operating with producer gas, the consumption has already been brought down to $\frac{8}{10}$ of a pound of coal per I.H.P. per hour, which is less than one-half that required for the best steam-engine installations.

As to the steam turbine, which is in much favor just at present, I might add that this type will always require more steam than a good reciprocating steam engine, and it is also a question whether it will be found satisfactory for use at sea in heavy weather. In my opinion it will never come into serious competition with the reciprocating steam or gas engine. The gas turbine might, however, be more highly recommended, as its character is such as to permit of construction in lengths more suitable for marine practice than its steam competitor.

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Figure 5, which represents an average diagram for a non-condensing steam engine, shows an efficiency of 7½ per cent. obtained from the coal in an ideal frictionless machine.

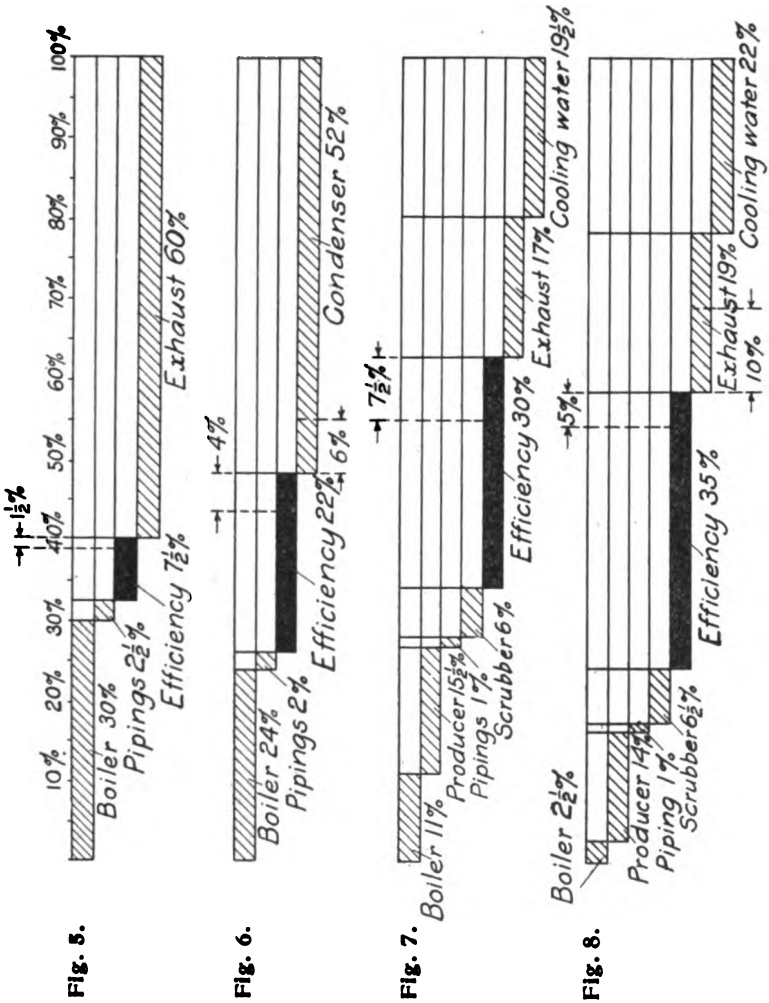

Fig. 5. Fig. 6. Fig. 7. Fig. 8.

Assuming a mechanical efficiency of 80 per cent., this amount is reduced to 6 per cent. The loss in an ordinary boiler may be taken as 30 per cent., with a further loss of 2½ per cent. in the piping and 60 per cent. in the exhaust, thus making the

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old type of engine a very uneconomical machine. By the selection of a better type of boiler, the use of heaters for the feed water and superheaters for the steam, the boiler loss may be reduced to about 24 per cent., as shown in figure 6, which represents a good modern condensing steam engine, having a loss of 52 per cent. in the exhaust. For this type the efficiency may reach as high as 22 per cent., from which must be deducted about 4 per cent. for mechanical losses, leaving a maximum efficiency of 18 per cent., a figure which has never yet been reached in marine practice, as the reversible type of engine required for this work can not equal the best stationary machines.

For comparison with the above, a similar diagram, figure 7, is given for a "pressure-producer" gas plant. As is generally known, the losses in such a plant are principally in the boiler used to furnish high-pressure steam for the producer and in the scrubber, in which the gas is cooled before going to the holder, from which it passes to the engine. The loss of heat units in the auxiliary boiler is as high as 11 per cent., in the producer about 15½ per cent., in the piping 1 per cent., and 6 per cent. in the scrubber. In a good, modern two-cycle engine there may be carried away in the exhaust as much as 17 per cent. of the heat units, and in the cooling water 19½ per cent., thus leaving for the plant a thermal efficiency of 30 per cent.

It will be noted that a four-cycle engine is just as well adapted for use in connection with a pressure-producer plant; but, from my experience with these engines, I would advise the adoption of the two-cycle type for marine purposes, in spite of a possible higher thermal efficiency to be obtained from the other, when used with a suction producer.

In the case of the two-cycle engine there is required an auxiliary compressor plant for the air and gas in which the efficiency loss amounts to about 7½ per cent., thus bringing the total efficiency down to 22½ per cent., as shown in figure 7.

Figure 8 is a diagram showing the distribution of thermal units in a modern stationary suction-producer plant of large size. In this case steam of high pressure is not used, but

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there is a vaporizer in which the loss amounts to 2½ per cent.; the loss in the producer is reduced slightly, amounting to 14 per cent., as compared with 15½ per cent. in the pressure type. With a loss of 7½ per cent. in the piping and scrubber, 19 per cent. in the exhaust and 22 per cent. in the cooling water, there still remains 35 per cent. of the calorific value of the fuel for the engine; allowing 5 per cent. loss for the engine, we have a brake horse-power efficiency of 30 per cent. In my opinion these figures are such as to warrant the building of marine gas engines.

The following calculation shows the heat-balance scheme for a pressure-producer plant using fuel with a thermal efficiency of approximately 13,100 B.T.U.s:

Coal charged per hour in producer, pounds	225
boiler, pounds	25
total, pounds	250
Total heat units charged in plant per hour	3,275,000
boiler per hour	327,500
furnished by steam per hour	181,600
lost in boiler per hour	145,900
charged in producer per hour	2,947,500
added to producer by steam	181,600
in producer	3,129,100
The gas obtained from the 225 pounds of fuel was approximately 17,000 cubic feet, with a calorific value of 150 B.T.U.'s, thus giving in heat units	2,550,000
Loss in producer was thus	579,100

The efficiency loss in the producer was, then,

$$\frac{579,100}{3,275,000} = 17.8 \text{ pr. ct.}$$

The efficiency loss of the boiler $= \frac{579,100}{3,275,000} = 4.5$ pr. ct.

Total loss in plant, ... $= 22.3$ pr. ct.

This may also be arrived at in the following manner: There was obtained per pound of fuel in the producer 75.6 cubic feet of gas, or in the plant, 68 cubic feet, having a calorific value of 150 B.T.U.s, thus giving approximately 10,200 B.T.U.s, whereas the value of the fuel was 13,100 B.T.U.s, which shows a loss of about 22.2 per cent.

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For gas with a value of 140 B.T.U.s the efficiency is 72.6 per cent. as compared with 77.8 in the above case.

From figures 5 to 8 it will be seen that there is no very great difference between a good boiler plant and a good producer plant. The gas engine, however, should be preferred not only on account of its superior economy but for the reason that it is safer, as in this case we do away with a system of piping carrying steam of a very high and dangerous pressure. The supply of coal is easier in the case of the producer than in that

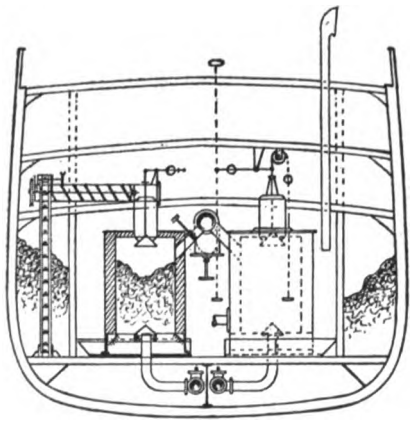

Fig. 9.

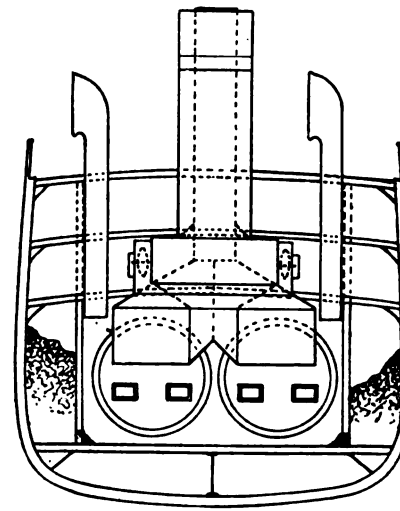

Fig. 10.

of the boiler and, as will be seen by comparing figures 9 and 10, the form of producer is such that it can be installed on board ship without greater difficulty.

In figure 11 is shown the general arrangement proposed for the engine room.

(Some further points as to the installation of large gas engines in the case of railroad locomotives will be found in an article by the writer which appeared in the "Scientific American" of February 4, 1905.)

In connection with existing installations of small internal-combustion engines I would call your attention to figures 12

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and 13, showing the plan and elevation of a Fairbanks-Morse 2-H.P. gasoline engine for a launch, the details of which will be seen in figures 14 to 16. It is of the two-cycle type, and has the advantage of no valves on the cylinders, a feature which is of such importance that I would propose to accept this general principle for engines of the largest type. There is only one automatic valve for the air suction.

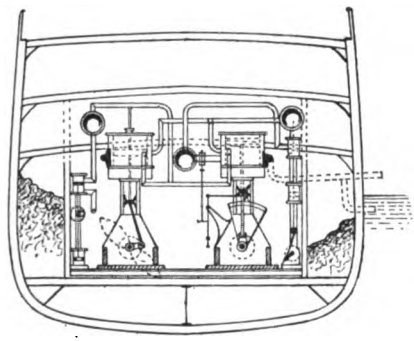

Fig. 11.

That the reversing shaft may be avoided has been demonstrated in the engines built by the Standard Motor Company and others. A reverse motion may be obtained by advancing the point of ignition so as to give an early back pressure on the piston in the proper direction; by shifting the valve motion; by the use of an auxiliary air compressor plant; or by the proper distribution of the explosives in the various cylinders.

In connection with the small engine just referred to, it may be of interest to examine the cylinder details as shown in figure 17. The exhaust opening is located near the lower end, with the mixture inlet a little higher up, at a; the piston in its lower position, b, opens the exhaust as well as the inlet, thus scavenging the cylinder and at the same time filling it. This type is very satisfactory even though not so economical as larger engines working on the four-cycle principle, but the latter are more or less complicated owing to the various valves and the mechanism necessary to operate them.

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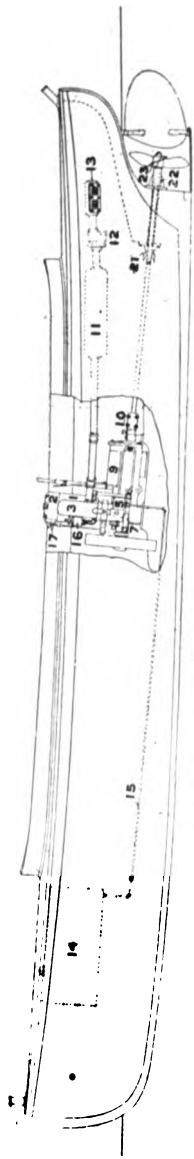

1. Cylinder.
2. Cylinder Head.
3. Heater.
4. Throttle Valve.
5. Float Reservoir.
6. Oil Valve.
7. Circulating Pump.
8. Sea Cock.
9. Reversing Gear.
10. Coupling.
11. Exhaust Pot.
12. Exhaust Muffler.
13. Exhaust Elbow.
14. Oil Tank.
15. Oil Feed Pipe.
16. Lubricator.
17. Electric Igniter.
18. Battery.
19. Switch.
20. Spark Coil.
21. Inboard Stuffing Box.
22. Outboard Stern Bearing.
23. Propeller Wheel.

Fig. 12.

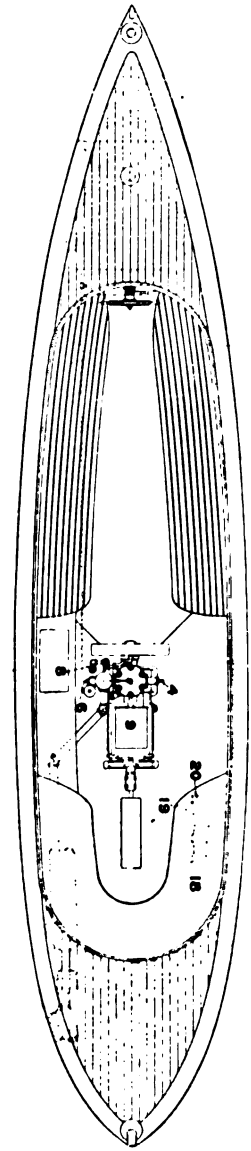

Fig. 13.

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It may be asked why, in view of the greater economy of the four-cycle engine, I should propose the two-cycle type for use on board ship. It is because the production of *poor* gasses can be easily carried out in modern producers and these gases are suitable for this type of engine, while gasoline and such fuels are too expensive for use; and then, in order to save weight, the engines will necessarily have to be made double acting. The gas under pressure is obtained from the producer direct, and the air required for scavenging may be compressed to the few pounds required by any convenient auxiliary apparatus.

Figure 18 shows the framing, drawn to scale, for a 10,000-H.P. engine, from which it will be seen that the lower part may be open, as in the case of a steam engine, and the cross-heads, crank pins, piston and rod-cooling devices, main bearings, etc., will be easily accessible. It will be noted that this engine is of the same size as a steam engine of corresponding power, its height being 36 feet and length 34 feet.

A proposed method of constructing the large cylinders for such high-pressures is shown by figure 19. A good proportion would be cylinder diameter, 60 to 70 inches, and stroke, 72 inches. The exact diameter of cylinders would depend, of course, upon the quality of the gas to be used. With eighty revolutions per minute this would give a piston speed of 960 feet. It should be remarked that for a steam engine of this same power there would be required a low-pressure cylinder with a diameter of 112 inches. There is, however, one point which might at first appear to place the gas engine at a great disadvantage; it is the enormous pressure brought to bear on the crank pins and main bearings by the force of the explosions. With a piston area between 2,827 and 4,071 square inches, and a pressure of 400 to 450 pounds per square inch, the resulting pressure on the bearings would be between 580 and 700 tons.

With a plentiful supply of water, which may be obtained from the sea, trouble from this source can be avoided. It will not be necessary to provide fresh or filtered water for cooling

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purposes, sea water being used, provided proper arrangements are made for cleaning out the various passages.

The next question of importance is to get rid of the heavy

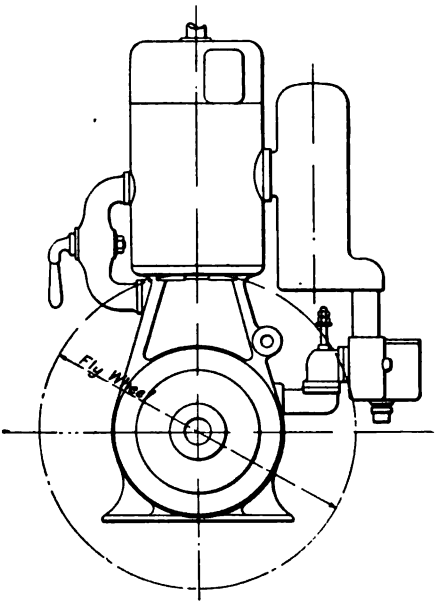

Fig. 14.

cylinders. If made of cast iron its thickness would have to be about 4½ or 5 inches, and for cast steel 2½ to 3 inches. There can be obtained from existing mills seamless-rolled cylinders, with diameters of 6 feet and over, in lengths of 10 feet. It is such cylinders, of nickel-steel, that I would propose to use for the large engines. Even if it should be found necessary to make them in two sections, the thickness in such cases would not be more than 1½ to 2 inches.

From figure 19 it will be seen that the openings for inlet and exhaust come at the end of the stroke, when the pressure in the cylinder is very low, being not more than about 60 pounds, so that no trouble need be experienced with the connections at these points. The cylinder heads may be tied together by rods, as in the case of an hydraulic press.

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By using specially operated gas valves, as shown by the dotted lines in figure 19, it would be possible to employ air

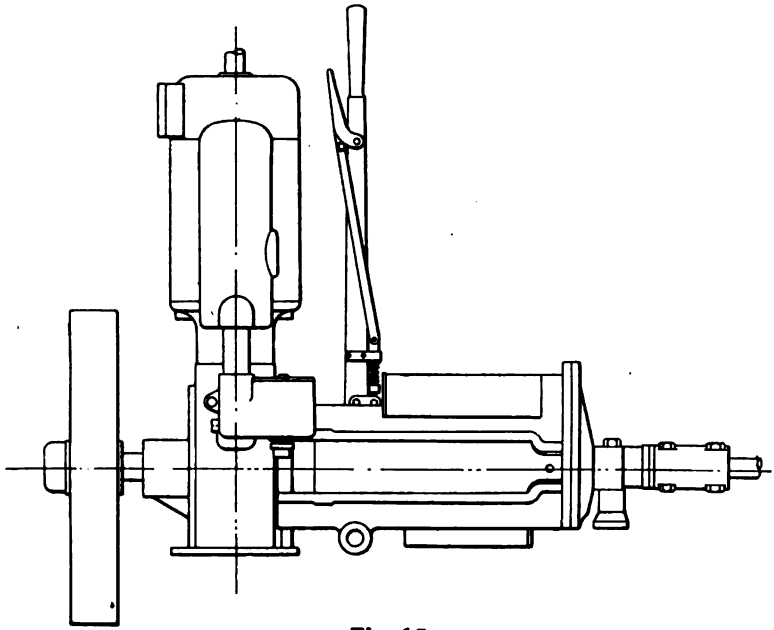

Fig. 15.

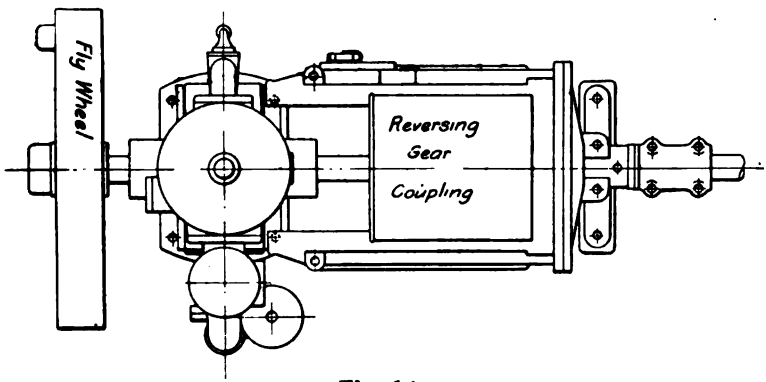

Fig. 16.

alone for scavenging and thus effect a saving in fuel.

The gas engine itself will be found heavier than a steam

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engine of the same power, but as the condensers with their fittings will not be required the final result may be an actual saving in weight.

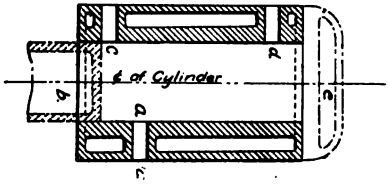

Fig. 17.

A note might be added concerning the rapid development of the so-called "waste-heat engines." It has been demon-

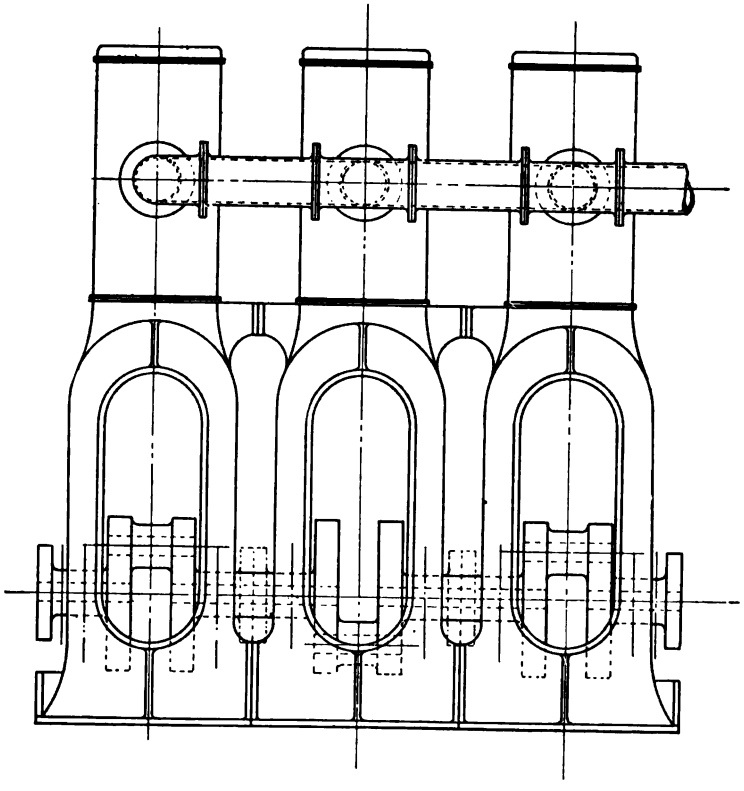

Fig. 18.

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strated that a saving of 6 per cent. may be obtained in the case of large condensing steam plants, as indicated on figure 6, and it is expected that in the case of gas engines this will amount to as much as 10 per cent. (see figure 8), the waste heat being utilized for the electrical operation of the auxiliary machinery, for heating the vessel and other similar purposes.

In a steam engine installation of 10,000 H.P. it is probable that there would be installed five boilers, each with a capacity of 2,000-H.P., which, in the case of cylindrical boilers, would weigh, including water, smokepipes and fittings, about 220

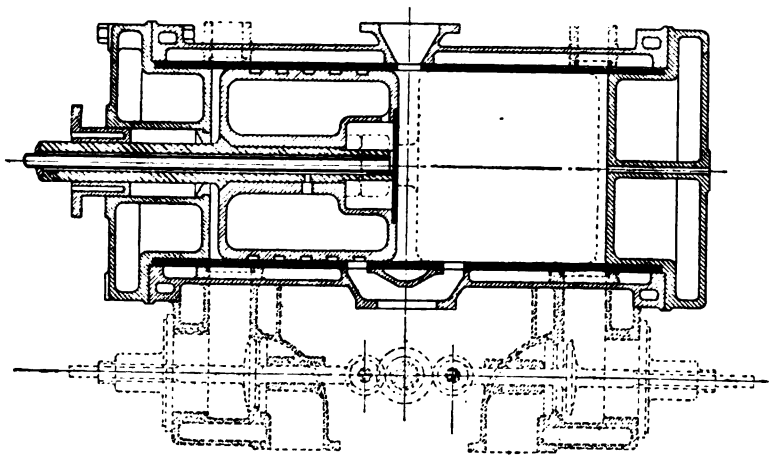

Fig. 19.

pounds per H.P.; though in the case of certain types, such as that invented by Professor Schutte, of Danzig, this weight has been reduced to 150 pounds. Taking 200 pounds as an average, there would then be for a steam plant a total of 2,000,000 pounds, or 1,000 short tons, in addition to the engine weights. The grate area would amount to about 750 square feet and the floor space required would be about 1,500 square feet.

For a producer plant it is safe to count on 10 square inches of grate per H.P., though this may, in large installations, be reduced to 8 square inches or even less, thus giving a total of 700 square feet. For a continuous supply of gas of a good

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quality it would be advisable to divide the plant into a number of small units. I would propose that ten producers, each with a capacity of 1,000 H.P. and 70 square feet of grate area, be adopted. The inside diameter of each would then be about 10 feet, which would correspond to an outside diameter of 12 feet, and would require for the whole plant a floor space of 1,140 square feet, as compared with 1,500 in the case of the boilers.

As to the question of weight, this is also in favor of the producers; each would weigh about 125,000 pounds, of which amount 77,000 pounds would be in fire-brick, and the remainder, 48,000 pounds, in the steel shell and fittings. This would give a weight of 125 pounds per H.P., or a total of 625 short tons, and show a saving of about 375 short tons.

For the operation of the plant there will be required no skilled fireman and a smaller number of engineers, and the probability of serious accidents will have been lessened.

In a system for which the writer has secured patents the scrubber, heretofore attached to each producer, has been omitted, only one central cleaning plant being installed and no gas holder used, even in the case of a pressure producer, thus making the installation one particularly well suited for use on board ship.

Every naval officer will readily appreciate the great advantage to be obtained by avoiding the use of smoke pipes, and thus reducing the chances of the vessel being discovered at great distances by the clouds of smoke, as at present.

For vessels of the Navy the space and weight saved may be utilized for an increase in the armor and armament or for additional coal supply. With the same coal capacity as heretofore, the saving of about one-third in the coal consumption will give a much greater steaming radius.

There will be no competition between this type of installation and one using a liquid fuel, as the latter is too expensive as well as dangerous for use on shipboard.

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FUEL FOR SHIPS OF WAR.

ITS ENGINEERING AND MILITARY RELATIONS.

By Lieutenant H. C. Dinger, U. S. N., Member.

The comparative efficiency of vessels of war, the offensive and defensive qualities being equal, will depend upon the endurance and steaming radius.

The endurance will depend upon the reliability and durability of the machinery, which qualities are obtained in the highest degree by proper design, careful arrangement and installation, the use of good material and workmanship, and last, but not least, proper care and management.

The steaming radius will depend upon, 1. Bunker capacity; 2. Quality of fuel; 3. Efficiency of devices used for converting the energy of the fuel into that of the steam; 4. Efficiency of the machinery for converting the energy of the steam into propulsive effect.

COAL BUNKERS.

The bunker capacity is determined by the space available. To be efficient it must provide facility for stowing and completely filling all bunker space, and means for readily getting the coal to the furnaces.

Facility for stowing.—The primary object in carrying coal is for use as fuel in the boiler furnaces. This fact should not be lost sight of in arranging the coal bunkers of a naval vessel. From numerous examples in existence an uninitiated observer might be led to believe that fuel-carrying ability was the last thing considered in the design and arrangement of the bunkers.

It is true that coal protection has a certain amount of military value, but the knowledge of this fact should not lead us

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into dangerous paths. The idea of coal protection has led to grave errors; one of which is the designing of coal bunkers so that the coal, when once stowed, is placed in a position well adapted for protection; but, owing to their inaccessibility, the bunkers are practically useless for carrying fuel for the boilers. The bad effect of trying to use coal for protection manifests itself in having bunkers fitted in narrow and irregular spaces or in small pockets from which the fuel can neither be taken to the fires nor stowed from deck except with great difficulty.

Fuel is being constantly consumed, and there is thus present the liability of the bunkers being empty when the protection is needed. Protection by fuel is thus in any case an uncertain and indefinite quantity upon which dependence can not be placed.

The primary purpose of coal on board a vessel should be to generate power, and the primary idea in the arrangement of coal bunkers must be that of readily getting the coal to the bunkers and from the bunkers to the furnaces in the easiest possible way. To this end the major portion of the coal should be in large bunkers, which are close to and have the shortest and most direct communication with the fire room.

Bunkers should be few in number and of large capacity rather than great in number and of limited capacity. With fewer bunkers the number of chutes, scuttles, doors, etc., is very much less, thus avoiding considerable weight and complication. Large bunkers are also more easily stowed, while with small ones this is very difficult.

The only limitation placed upon the size of bunkers on board men-of-war should be such that the flooding of any one would not heel the vessel to a dangerous degree. Necessary strength can be obtained by fitting stanchions, stringers, beams, etc.

INTERNAL ARRANGEMENTS OF BUNKERS.

To facilitate stowing there should be provided: A sufficient number of *chutes* as large as the structural limitations will allow, located so that the coal will most easily distribute itself

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to all parts of the bunker. Such chutes should lead direct to the bunkers without intermediate passageways, twists or turns.

Proper *escape doors*, so located with reference to the chutes that the least amount of vacant space is left in bunker, yet still providing a means of escape for the trimmer.

Proper *ventilation*, in order that the coal trimmers may not be required to work in excessive discomfort.

To facilitate the taking of coal from the bunkers there should be—

Platforms from which the coal can be shoveled.

Trolleys and *switches* for transporting coal buckets from distant points.

Doors, *scuttles*, *hatches*, etc., must be so designed as to be easily opened and closed and to withstand rough handling.

EMPLOYMENT OF MECHANICAL MEANS.

Mechanical means for bringing coal from bunkers are in all probability not applicable to the conditions on board ship, chiefly on account of the weight and the space necessary for their installation: the space required for such installation being an encroachment upon the bunker capacity.

If mechanical appliances can be devised to do such work, which weigh less and occupy less space than men doing the same work, then there may be some reason for such an installation; at present no such devices are in sight.

QUALITY OF FUEL.

A first impression might give the idea that the fuel which contains the greatest number of heat units per unit of weight would be the most desirable. This is, however, not necessarily true. Other things must be considered, among which are:

1. Safety.
2. Facility for handling and stowing.
3. Facility and ease with which combustion can be obtained and maintained.
4. Maximum evaporative result that can be obtained with a unit weight of boiler of a type suitable for marine installation.

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5. Availability and facility of supply.

If these items are considered with regard to oil fuel, items 1 and 5 are of smaller value than with coal. Item 2, however, is much greater with oil.

Liquid Fuel.—The advantage of the use of liquid fuel, aside from all question of adequate supply or cost, may then be determined by properly considering its bearing on the above items.

Kinds of Coal.—Items 1 and 2 are practically alike for nearly all kinds of coal. Anthracite coal offers more advantage by reason of closer and cleaner stowage.

Under item 3, "Facility with which combustion can be obtained and maintained," the kind and character of the coal should become a matter of thorough consideration.

Coals are relatively efficient as follows:

(a) In as far as they show a small percentage of ash.

(b) In that they do not clinker.

(c) In that they are not likely to produce a secondary combustion.

(d) In that they are not ground to powder, which will allow large quantities of fuel to drop through the grate.

Item 4 is one that very often may not be carefully considered. Other things being equal, that fuel is best from which a certain evaporative result per unit weight can be obtained with the least weight of boiler. Fuel that would require added weight to the boiler installation in the way of special furnaces or other appliances will thus have a severe handicap.

AVAILABILITY OF SUPPLY.

Any fuel for naval vessels must, in general, have an available supply throughout the probable seat of operations for the vessel; otherwise her field of action is limited. Naval vessels, in general, must have the whole world for their field, and until a proper available supply of any particular fuel can be assured in all portions of the globe, such special fuel can not be advantageously used for naval purposes. With merchant vessels plying in a definite locality or on a definite run

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this is altogether a different question. For such naval vessels whose sphere of action is confined to a particular locality, fuels available in that locality and which have special advantages ought to be used.

Efficiency of conversion of the heat of fuel into steam will depend upon the following points:

Firing.—This at once brings up the question of hand firing or mechanical firing. Mechanical firing is successful on shore, but on board ship its limitations are such, especially on a man-of-war, that the probability of this means being used for burning coal is yet far distant.

The disadvantages are:

(a) *Weight.*—The added weight is much more than the weight of the additional men required for hand firing.

(b) *Space.*—The space required is more than that necessary for hand firing.

(c) *Adjustment for raising speed by increasing rate of combustion.*—This is very difficult, practically, with any mechanical system. Mechanical stokers do not work successfully at any where near the rate of combustion of 50 pounds per square foot of grate, this being the rate often necessary on account of limitations of space.

(d) *Complication entailed.*—The fittings of the mechanical stokers require considerable care and attention.

Hand firing.—Matters that will affect the efficiency of hand firing are:

(a) *Length of grate.*—Grates should not be over 6 feet 6 inches long; a longer grate cannot be properly fired for more than a short period. Attempts at firing long grates result in making the rear of grate inefficient, since this part cannot be thoroughly cleaned, nor can the fuel be properly broken up.

(b) *Height of fire-doors and the general ease with which they can be handled, slope of grate, etc., and iis relation to direction of passage of gases.*—Each of these matters, if not properly designed to meet attending conditions, will detract from efficient firing.

(c) *Depth of fire room.*—The depth must be such that slice

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bars can be handled with ease. The space in wake of furnaces must be absolutely clear, so that firemen run no danger of striking their hands against stanchions, ladders, etc.

(d) *Ventilation and temperature of fire rooms.*—The fire room for natural draft should provide a good supply of fresh air and ample means of escape for the heated air. For forced draft, discharges should be led so that the air is not likely to stir up coal dust.

(e) *Skill of firemen and system of firing employed.*—This, of course, is a feature of overwhelming importance. Another matter affecting the firing is the amount of ashes and arrangements provided for handling same. If these arrangements are bad, the ashes will interfere to a very considerable extent with the efficiency of the firing.

HEATING SURFACE.

This is efficient or non-efficient, from two main causes:

1. Position in relation to the current of gases and the distance the gases pass along the surface. Heating surfaces in which the circulation of water is at right angles to the current of gases are probably the most efficient.

2. Condition of heating surfaces as to cleanliness both inside and out. Dirt on inside of tubes is to be avoided by having surfaces so situated that dirt deposit is not likely to lodge on the heating surface, and by making such provision that dirt, when it does collect, can be readily removed.

Soot collecting on outside of tubes is to be avoided by the absence of lodging places and by providing for the cleaning of surfaces while boiler is steaming.

Proper consideration of these matters will increase the efficiency of the boiler to a very large extent, a fact which boiler manufacturers in many cases seem unable to grasp until after very costly experience.

The real relative steaming efficiency of any type of boiler can be determined only after long runs at full speed. The 4-hour trial trip is no criterion of service-steaming efficiency of a boiler, nor of its real economy and endurance.

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The evaporative result of a boiler should be stated in boiler efficiency, that is, the per cent. the actual heat of steam generated bears to heat of the coal burnt. So much evaporation per pound of coal, without stating the kind and heating value of the coal, does not give a definite or fair comparison of boiler performance.

For purposes of proper comparison, there ought to be a standard pound of coal containing say 14,500 B.T.U. Then all results could be worked from this basis, and there would be some chance of a real unprejudiced and clear comparison of different tests.

EFFICIENCY OF MACHINERY FOR PROPULSIVE EFFECT.

This will depend on:

1. Greatest effective power developed in engines per unit of steam used. Economy of engines—this is perhaps the most potent factor of all; steam consumption of engines varying from 30 pounds to 12 pounds of water per I.H.P.

2. Most efficient use of power for propulsive effect (efficiency of propellers). By careful design an efficiency of nearly 70 per cent. can be obtained; in practice much lower efficiencies are found.

3. Least resistance at speed per ton of displacement for any displacement (efficiency of hull). This will depend on original lines to give least resistance at a certain displacement.

Other matters affecting this are: Smoothness of the surface of hull, extent of marine growth, frequent docking, efficiency of anti-fouling paints, sheathing, etc. These are all matters a full discussion of which would in each case involve a lengthy essay, though exact data on many of the points are not available and have not been determined.

It will thus be seen that a great many matters, not ordinarily considered as having any bearing on the question of fuel, have a very considerable effect on the coal endurance of a naval vessel. The following is a resumé of these items:

1. Bunker capacity in weight and space.
2. Kind of fuel.

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3. Stowage of bunkers and facility for doing so.
4. Getting coal to fires and means for so doing.
5. General character and arrangement of fire rooms.
6. Skill of firemen and system of firing.
7. The detail and arrangement of furnaces.
8. Characteristics and arrangement of heating surface.
9. Arrangement provided for baffling gases.
10. Provisions for cleaning heating surface.
11. Evaporative efficiency of boiler as a whole.
12. Steam economy of engines and auxiliaries.
13. Mechanical efficiency of engine and shafting.
14. Efficiency of propellers.
15. Propulsive efficiency of lines of hull.
16. Condition of bottom.
17. Fuel used for purposes other than power for propulsion.

Any one of the above items and various others may have a very considerable effect upon the real steaming radius of a vessel. From this it should be seen that a calculated steaming radius based on bunker weight or space is most likely to be very misleading if some of the numerous other items vary. It can also be readily seen that steaming radius depends more on the character and design of boilers and machinery than it does on mere bunker capacity.

METHODS OF SUPPLYING FUEL.

The methods of supply must be viewed from the condition of affairs, not in ordinary time of peace, but in time of actual hostilities. The fallacy of building and preparing war material on a peace basis and for peace purposes is most harmful to military efficiency. To show its very harmful and misleading qualities it is but necessary to cite that it has been stated, in print, as a reason for *not* building a certain type of naval vessels "that they would be useful only in *time of war*."

In the matter of fuel supply, we want a system that will be useful in time of war. The question of its utility in time of peace should have but little weight.

The question of supplying a fleet with fuel in time of war is

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a problem of transporting fuel from the source of supply to a movable fleet. In order that this fleet be supplied, it must either go to a permanent station or a movable station must come to it. If the fleet goes to the station, the fleet's military value is lost during that time. If the movable station can go to the fleet, the only time taken away from the military duties will be the actual time needed for transferring the fuel.

It is obvious that to supply a fleet during a campaign the movable fuel station or collier is the most desirable.

OUTLYING COALING STATIONS.

In time of war outlying coaling stations, unless protected at a practically prohibitive cost, are almost as useful to the enemy as to friends. To be of any real value they must be very numerous and located all over the world. The amount of coal stowed must be excessive, for an adequate supply must be at each station.

Any system of outlying coaling stations will require an enormous force for their proper protection and upkeep, and for supplying these stations colliers are also needed.

The fighting ships will, however, at various times have to return to a refitting point, which must be a properly protected place. At such a station coal must also be obtained. The number of vessels likely to require coaling at a certain point can only be known a short time before the fuel is actually needed; hence a permanent land coaling plant that is not susceptible to varieties in output is not and cannot be the desired appurtenance. The coaling plant must be one susceptible of expansion or contraction for the varying needs. It therefore follows that our present expensive coaling plants, located as they are at various points along the coast, some fortified, some not, some in conjunction with a refitting and revictualling place and some not, some located so that only small vessels can use them, are hardly what would seem the most desirable, and are certainly not really worth what has been spent on them. As a matter of fact, the type of coaling plants now built and building may be rightfully considered almost valueless as far as their

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utility for purely military purposes is concerned; that is, for the actual operation of coaling fighting vessels in time of war. Might not just a little of the money that is being spent on these plants be used for an investigation and consideration of the question of the necessary and best-suited type of coal-carrying vessel for use in time of war?

The coaling apparatus at a refitting station should be a floating apparatus, so that any draught vessel can use it. The grave difficulties of going up a narrow channel to get alongside a coal wharf are thus obviated. The capacity can be increased to suit the probable movements of the fighting vessels. The supply of coal can be more readily replenished by any cargo boat, and the actual cost of the apparatus for the same total capacity will probably be less.

The type of such a movable coaling plant is a large barge, built so that if necessary it can be towed to sea. The barge should be fitted with coal-hoisting gear and a power plant for operating it. There should also be on board an electric-lighting plant and living quarters for the necessary attendants. Such barges should hold several thousand tons of coal; they could be anchored in a harbor; the fighting vessels could go alongside to get their coal, and the supply colliers bringing coal to them could unload alongside. If the harbor has a commercial coal-supply depot, these barges could be towed to such depots and loaded direct.

Smaller lighters should be used in conjunction with the large floating stations, for transporting coal as required to any point in the vicinity.

COLLIERS.

The purpose of fleet colliers should be to take the fuel from the various sources of supply and bring it to the fleet rendezvous or movable station, wherever that may be, and to supply coal to vessels on blockade.

GENERAL CHARACTERISTICS OF COLLIERS.

Speed.—The speed should be such that their maximum is the speed at which the battleship fleet might steam in squad-

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ron. With our present battleships at 18 knots, a speed of 16 knots for colliers would seem to be the maximum desirable. In this connection it must be remembered that an extra knot above 16 requires a very large increase in power.

The displacement should be the largest compatible with a draught that will allow the vessel to come alongside the ordinary coal wharf of our harbors and to enter any ordinary harbor of importance. The draught, for these reasons, should not exceed 25 feet, and full load displacement would be limited to about 16,000 tons. With this displacement 10,000 tons of coal might be carried.

Every fleet collier should also carry oil and bulky stores, chiefly engineering stores, that require repeated replenishment. The securing of lubricating oil is even of more importance than coal.

The problem of supply could then resolve itself into—

1. Securing within various fortified harbors large storage barges, alongside of which men-of-war or fleet colliers can coal, and which are to be supplied by cargo steamers from the sources of supply. As supplementary to such floating stations, there ought to be a number of small coal barges which could be towed to vessels which might not find it convenient to come alongside of the large barge or which might not be able to do so.

2. A sufficient number of fleet colliers, specially designed for keeping up with a fleet at ordinary cruising speed.

3. Ordinary merchant colliers, chartered as needed, to transport coal to floating stations and to assist in bringing coal to the fleet when the resources of the regular colliers are overtaxed.

An adequate equipment of such fuel-supply facilities can be supplied at the cost of a few million dollars, which will be of real utility twenty years from now; yet, with all that has been spent in the last ten years on shore-coaling plants, we have hardly one that is of real practical use for coaling the fighting ship in time of war.

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THE LOVEKIN IMPROVED OUTBOARD LOOSE COUPLING.

By Robert S. Riley.

The type of outboard loose coupling illustrated in Fig. 1 has been designed with a view to overcoming some of the objections inherent in the sleeve type which has been so generally used to connect the propeller and tail shafts in our large naval vessels.

It is well known that the old sleeve type, as illustrated in Fig. 2, involves some difficult problems in manufacture and fitting. But its compactness, strength and neat appearance have been such factors in its favor that no attempt has been made to supersede it. The couplings illustrated here are designed for 16½-inch shafts, such as are used for battleships of the *Kansas* class.

Shop men particularly appreciate the difficulty in machining the interior of the old sleeve coupling to the degree of accuracy which is absolutely essential for its efficiency. The long taper fits are an objectionable feature, and so are the cotter keys which pass through both shaft and coupling. But the most difficult work is the accurate cutting and fitting of the long keyways. It takes a special rig to plane such keyways properly. One of the best devices for this purpose is a heavy cast-iron column passing through the coupling and carrying a tool post. The coupling is fixed to the planer platen. The bar slides in supports attached to the platen, but gets its motion from the planer head. But such special rigs for work of this size and accuracy are very expensive.

The fitting of the long keys in the interior of the sleeve is a job which can be appreciated only when one considers that to get at the middle of the sleeve a man must reach in a distance of 32¾ inches (nearly 3 feet), and this through a hole only 16½ inches in diameter at the outside.

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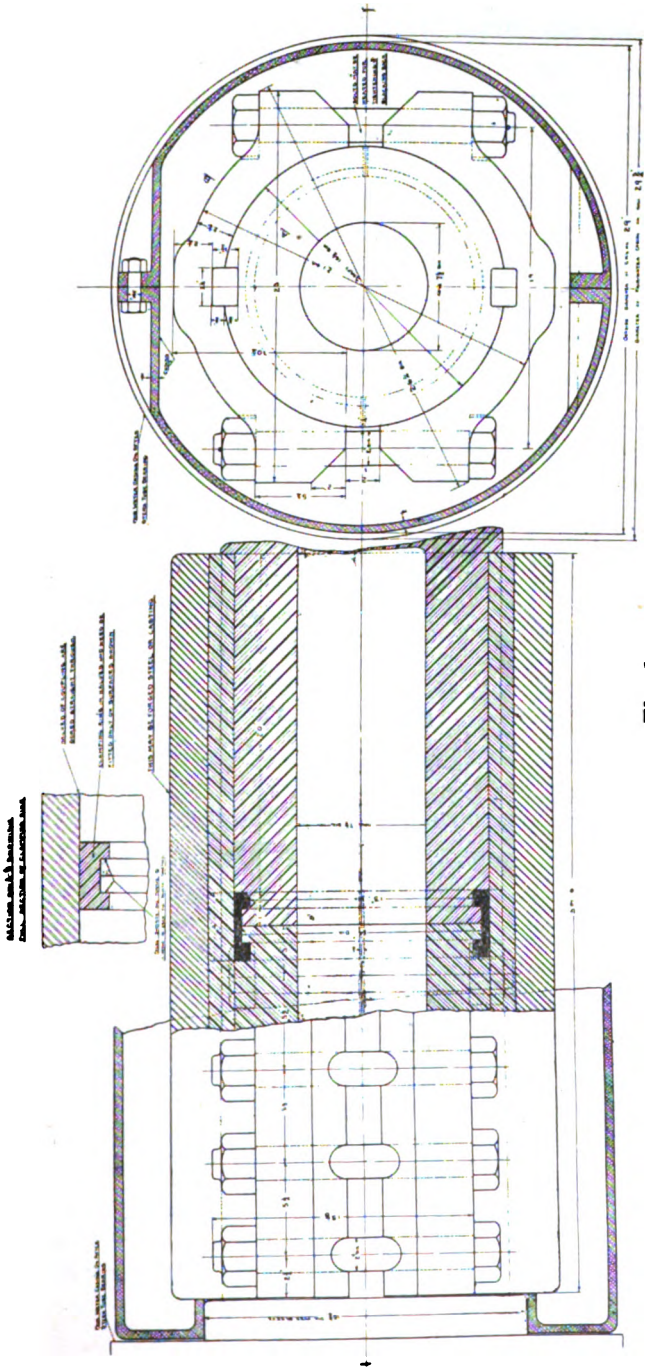

Fig. 1.

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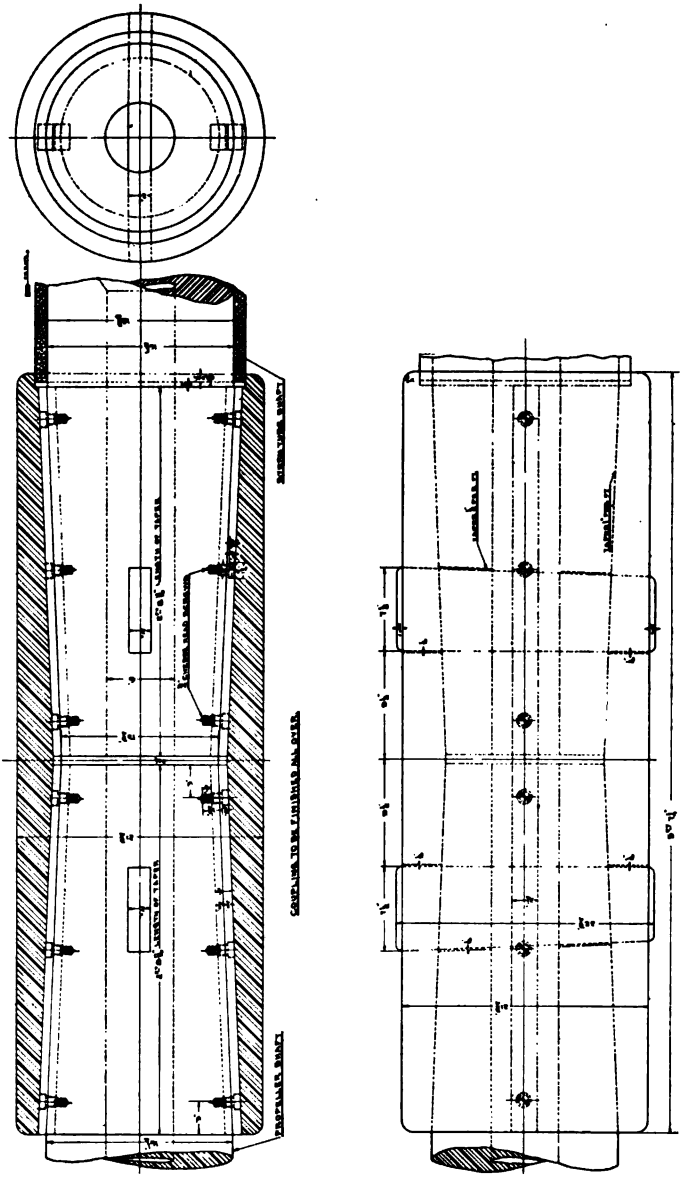

Fig. 2.

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Even after the coupling is fitted there comes the question of its usefulness for the purpose intended. A loose coupling should surely be easily removable. But it will be readily seen that a very little corrosion in the interior would render this design anything but "loose;" in fact, its removal might become a very serious problem. It would not be hard to imagine a critical emergency arising from this cause.

The new type of clamp coupling shown is the idea of Mr. Luther D. Lovekin, Chief Engineer of the New York Shipbuilding Company, of Camden, N. J. It is now being built for use on the battleship *New Hampshire*. The advantages to be gained by this design of coupling may be summed up as follows:

1st. There are no difficult problems in its manufacture. There is not a taper fit anywhere. The boring is straight and parallel. The keyways are planed straight through and may be done without special tools, any number at a time, depending on the length of the planer bed. The clamping ring need fit only on two inside surfaces.

2d. The keys can be fitted accurately in place so as to get their proper distribution of the load. The coupling is assembled in pieces of convenient size and in such a manner that there is no uncertainty as to fit; whereas in the sleeve type of outboard coupling the work has to be done inside at different angles, and it is difficult to try the keys in place.

3d. The propeller shaft, weighing about 15,000 pounds, does not have to be moved in the process of fitting the coupling and keys. The ends of the shaft may be brought together in proper alignment, and all fitting and clamping done without further movement of the shaft. With the sleeve type the shaft and coupling have to be moved at each test.

4th. The reduced section of the shaft is not subjected to any stress either of torsion or bending; the only service required of the reduced section is the thrust of the propeller, and that only when going astern.

5th. There is no hole cut through the shaft as required for the cotters in the sleeve type. The drawing effect of these

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cotters is uncertain, as the whole shaft has to be moved at each blow on the cotter, hence it is never known how closely the shaft fits the interior of the sleeve and keyway.

6th. The coupling is easily removed and replaced when necessary in ordinary service. The shaft need not be moved, and the parts of the coupling are of convenient size for handling. The clamping bolts can be readily expanded by the application of heat for the purpose of tightening or slacking back. Corrosion of the interior surfaces of the coupling and shaft cannot render the removal of the coupling impossible, as would be the case in the sleeve type.

7th. It is about 1,500 pounds lighter than the sleeve type, owing to reduced length. (This is 3,000 pounds in one ship.)

8th. It is of ample strength, as shown by calculations, and is safer than the sleeve type, owing to the fact that the stresses can be properly distributed by the accurate fitting made possible in its design.

It might be mentioned also that the design might be modified to suit the material to be used. For instance, if cast steel were used it might be desirable to change the form slightly, so as to have the coupling heavier at the center, where subject to the greatest torsion. The form shown is the simplest and plainest, so far as outside machining is concerned. In the *New Hampshire* the coupling is to be of forged steel.

To offset these advantages there are very few disadvantages to be urged. The increased volume necessary to clear the bolt heads and nuts is certainly no advantage; but as the diameter of the casing can still be less than that of the fair water to which it is adjacent, this is not a disadvantage.

Below are appended the calculations involved in this design. They show that the requirements are fully met in every particular.

CONDITIONS REQUIRED.

Horsepower $=16{,}500$ total $=8{,}250$ H.P. each shaft.

R.P.M $=120=N$.

Inch pounds per minute $=8{,}250\times 33{,}000\times 12$.

This represents the work imposed by the designed horsepower.

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Let M represent the resulting moment or torque on the shaft in inch pounds.

With this moment the work done by the shaft will be $2\pi NM$ inch pounds per minute.

Hence, $8{,}250 \times 33{,}000 \times 12 = 2\pi NM$.

Therefore, $M = \frac{8{,}250 \times 33{,}000 \times 12}{2\pi \times 120} = 4{,}335{,}000$ inch pounds.

Diameter of propeller shaft:

Outside diameter $= 16\frac{1}{2}$ inches.
Inside diameter $= 9\frac{1}{2}$ inches.

The coupling is to be designed so as to be at least equal to the strength of the shaft at every point.

Moment of resistance to torsion in hollow shafts

$$= .1965\, S \frac{(D^4 - d^4)}{(D)}.$$

Where S = stress in pounds per square inch.
D = outside diameter.
d = inside diameter.

$$D^4 = \overline{16\tfrac{1}{2}}^2 = 74{,}120.$$
$$d^4 = \overline{9\tfrac{1}{2}}^2 = 8{,}145.$$
$$D^4 - d^4 = 65{,}975$$
$$M = .1965 \times \frac{65{,}975}{16.5} \times S.$$
$$= 785\, S.$$

Hence $S = \frac{4{,}335{,}000}{785} = 5{,}522$ pounds per square inch in the normal section of the shaft due to torsion.

Thrust on Coupling.

Indicated thrust $= \frac{\text{I.H.P.} \times 33{,}000}{P \times R}$.

P = pitch of propeller $= 17.5$ feet.
R = R.P.M. $= 120$.

Indicated thrust $= \frac{8{,}250 \times 33{,}000}{17.5 \times 120} = 129{,}600$ pounds.

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For actual estimated thrust allow $\frac{2}{3}$ of 129,600=86,400 pounds.

To resist this thrust when going astern, fit the sectional ring as shown in drawing, so as to act as a clamp in holding the ends of the shafts together.

Find crushing stress imposed on the surface of the rings holding the sections of shaft together.

Outside diameter 14 inches—area	=153.94
Inside diameter 13 inches—area	=132.73
Net area=	21.21

From this subtract the amount cut out of clamping rings where they fail to meet, say $\frac{1}{4}$ inch $\times \frac{1}{2}$ inch $\times 2 = .25$ square inches.

$$\begin{array}{r} 21.21 \\ .25 \\ \hline \end{array}$$

Area subject to crushing=20.96, say 21 square inches.

Hence crushing pressure $= \frac{86,400}{21} = 4,120$ pounds per square inch.

This is the same both in ring and shafts.

For shearing stress, consider the root of rings on shaft=13 inches diameter—circumference=40.84.

Deduct $\frac{1}{2}$ inch for open space between ends of clamping ring, leaving 40.34 inches.

Length of 1 inch depth=40.34 square inches.

Hence shearing stress $= \frac{86,400}{40.34} = 2,090$ pounds per square inch.

For tensile stress on ring when the shafts are trying to separate there is no need for calculation, as it is obviously much less per square inch than either the crushing or shearing stresses.

Note.—From calculations later, on the clamping effect of the bolts, the resistance due to surface friction=92,000 pounds. This is greater than the estimated thrust=86,400 pounds.

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Hence the above stresses on the rings can never be attained and the rings may not be used at all.

Stress on Keys.

Twisting moment=4,335,000 inch pounds.

Keys are to resist this torque.

Use two keys $1\frac{1}{2}$ inches deep=$\frac{3}{4}$ inch deep in shaft and coupling.

Diameter of center of keys=$16\frac{1}{2}$ inches and radius (or arm) =$8\frac{1}{4}$ inches.

Shaft keyway is subject to greatest stress because the arm of its moment is less.

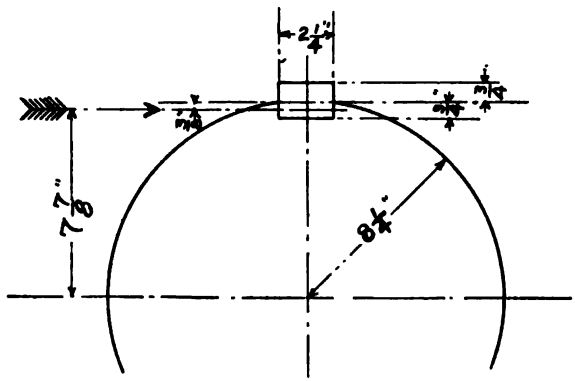

Fig. 3.

Radius of center of crushing forces in shaft keyway=$8\frac{1}{4}$ inches—$\frac{1}{2}$ of $\frac{3}{4}$ inch=$7\frac{7}{8}$ inches=7.875 inches, assuming forces concentrated at center of bearing surface of keyway. See Fig. 3.

$$\text{Pounds pressure at that arm} = \frac{\text{moment}}{\text{radius}}.$$

$$= \frac{4{,}335{,}000}{7.875}.$$

$$= 550{,}000 \text{ pounds for both keys.}$$

$$= 275{,}000 \text{ pounds for each key.}$$

Length of key=20 inches in each shaft.

Area of side of keyway in shaft=$\frac{3}{4}$ inch $\times$ 20 inches.

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Hence, crushing stress $=\frac{275,000}{\frac{3}{4}\times 20}$

$=18,320$ pounds per square inch.

Width of key $=2\frac{1}{4}$ inches.

Area of two keys to resist shear $=2\frac{1}{4}$ inches $\times 20$ inches $\times 2$.

Arm of shearing stress $=8\frac{1}{4}$ inches.

Hence, shearing stress $=\frac{4,335,000}{2\frac{1}{4}\times 20\times 2\times 8\frac{1}{4}}$

$=5,840$ pounds per square inch.

Note.—These stresses on keys are decreased by the clamping effect of the loose coupling. See other calculation later.

CLAMPING POWER OF 16 BOLTS 2 INCHES DIAMETER.

At 5,000 pounds stress per square inch at root of thread, the pressure due to one bolt $=11,500$ pounds.

Hence the pressure of 16 bolts $=16\times 11,500=184,000$ pounds total.

This is over an area 44 inches long and $16\frac{1}{2}$ inches diameter.

Width $=\frac{\text{circumference}}{2}-2$ inches $=\frac{51.8}{2}-2$ inches $=23.9$ inches (allowing for space where the halves of coupling fail to meet).

Pressure per square inch $=\frac{184,000}{44\times 23.9}=175$ pounds.

At this pressure the coefficient of friction for wrought or cast steel will be at least .25 (friction of rest—unlubricated).

Hence, resistance of the entire surfaces of the coupling $=184,000\times .25\times 2=92,000$ pounds. This is along the surface of the shaft.

Hence, the moment $=92,000\times 8\frac{1}{4}$ inches radius,

$=759,000$ inch pounds.

This may be subtracted from the moment for which the keys were designed $=4,335,000$

$759,000$

Leaving $3,576,000$ inch pounds.

Pressure of this moment at radius $7\frac{7}{8}$ inches (see Fig. 1)

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$$=\frac{3,576,000}{7.875}=454,000 \text{ pounds.}$$

Hence, pressure per square inch on the keys would be

$$\frac{454,000}{2\times 20\times \frac{3}{4}}=15,130 \text{ pounds per square inch (say 15,000),}$$

and shear would be $\frac{3,576,000}{8\frac{1}{4}\times 2\frac{1}{4}\times 20\times 2}=4,810$ pounds.

The total effective thrust of the engines is 86,400 pounds (approximately); hence the friction might be more than sufficient to dispense with grooves entirely.

Moment of resistance to torsion of loose coupling.—Section at end of keys:

Area = as measured by planimeter from drawing = 159.04 square inches.

Outside diameter (approximately) = 21 inches = D.
Inside diameter (approximately) = $16\frac{1}{2}$ inches = d.

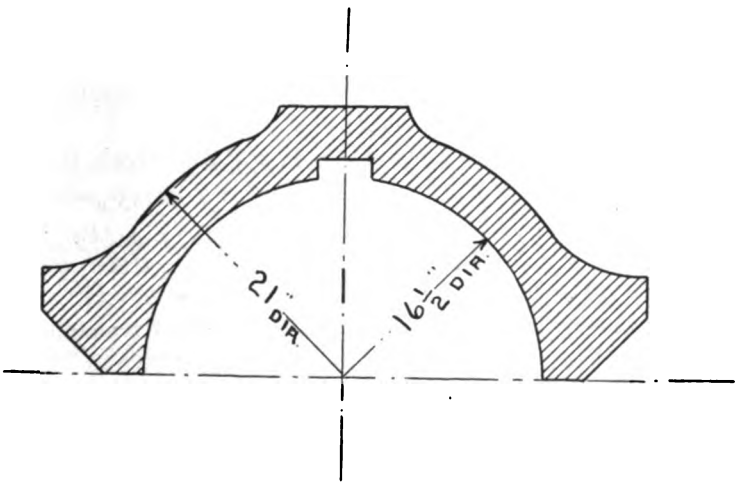

Fig. 4.

Assume that the forces may be considered acting at the center line of a longitudinal section, the same as for a thin shell. See correction later.

The radius of which is a mean between the inner and outer radii $=\frac{1}{4}(D+d)=\frac{37.5}{4}$.

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Area of transverse section under consideration = 159.04 square inches.

(Area of complete ring between similar diameters =

$$\frac{\pi}{4}(D^2 - d^2) = 132.5.)$$

Moment of the section to resist torsion = stress per square inch × area × arm = $S \times 159.04 \times \frac{37.5}{4}$.

To allow for the inaccuracy of this method between these diameters, multiply this moment by .905, as calculated. See in note later.

$M = S \times 159.04 \times \frac{37.5}{4} \times .905 = 1{,}349$ "S" inch pounds.

Moment for the normal section of shaft = 785.S inch pounds.

Hence the section is $\frac{1{,}349}{785} = 1.72$ times as strong as the normal section of shaft to resist torsion.

= 72 per cent. stronger (but is required for stiffness and safety.)

Moment required for 8,250 H.P. = 4,335,000 inch pounds.

Stress in coupling to transmit this torque = $\frac{4{,}335{,}000}{1{,}349}$.

= 3,210 pounds per square inch.

Note.—Re factor used to modify moment of resistance to torsion.

Moment of resistance to torsion in hollow shafts.

$$M = \frac{\pi S}{16D}(D^4 - d^4).$$

$D = 21$ inches.

$d = 16\frac{1}{2}$ inches.

$$M = \frac{\pi S}{16}\,\frac{(120{,}361)}{21}.$$

$$= \frac{\pi S}{16} \times 5{,}731.5, \qquad (A)$$

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This is the correct moment of resistance for this shaft.

Now consider this shaft as a circular ring for purposes of comparison.

$$\text{Area} = \frac{\pi}{4}(D^2 - d^2).$$

Radius of center of gravity of longitudinal cross section

$$= \frac{1}{4}(D + d).$$

The moment of resistance to torsion by this method is therefore,

$$M = S \times \frac{\pi}{16}(D^2 - d^2) \times (D + d).$$

$$M = \frac{\pi}{16}(168.75 \times 37.5).$$

$$= \frac{\pi}{16} \times 6{,}328.125, \quad . \quad . \quad . \quad (B)$$

This is an approximation of the moment worked out in the same manner as assumed for the cross section of the coupling.

This shows that the latter method gives a value of $\frac{6{,}328}{5{,}731}$ times too great for the diameters chosen.

Hence, results on this method with similar dimensions must be multiplied by $\frac{5{,}731}{6{,}328} = .905$ in order to get the correct value of the moment of resistance to torsion.

To find diameter of hole required and have the shaft equal to original before keyways were cut out.

Radius of center of gravity of keyway $= 8\frac{1}{4}$ inches $- \frac{3}{8}$ inch $= 7\frac{7}{8}$ inches.

Area of keyway $= \frac{3}{4}$ inch $\times 2\frac{1}{4}$ inches $= 6\frac{3}{4}$ square inches.

Moment of resistance to torsion of this material cut out of these keyways will be approximately the moment of their shearing values $= 6\frac{3}{4} \times S \times 7\frac{7}{8} \times 2$.

This must be compensated for by a hollow shaft inside the original, of which $D = 9\frac{1}{2}$ inches.

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$$D^4 = 8{,}145.$$
$$d = ?$$

The moment of resistance of the shaft must therefore equal the above moment of the material cut out for keyways.

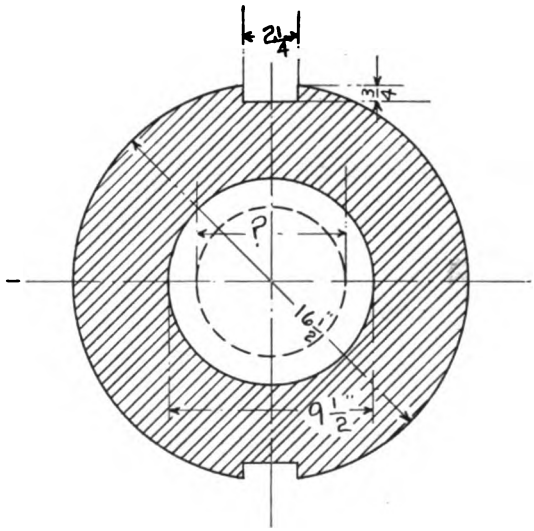

Fig. 5.

Hence, $\frac{\pi S}{16D}(D^4 - d^4) = S \times 6\frac{3}{4} \times 7\frac{7}{8} \times 2.$

and $d^4 = \frac{-6.75 \times 7.875 \times 2 \times 9.5 \times 16}{3.1416} + 8{,}145$

$= 8{,}145 - 5{,}140 = 3{,}005.$

$d = 7.704$ (say $7\frac{3}{4}$ inches).

Make diameter of hole $= 7\frac{1}{2}$ inches. (Ample.)

ACTION OF KEYS IN SPREADING THE COUPLING.

The driving pressure F on one side of the shaft key seat at radius $7\frac{7}{8}$ inches is 275,000 pounds, previously calculated.

Hence, resisting pressure R in the coupling at radius $8\frac{5}{8}$ inches $= 275{,}000 \times \frac{7\frac{7}{8}}{8\frac{5}{8}} = 251{,}000$ pounds.

These pressures both act against each other at arms of $\frac{3}{8}$ inch each.

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Therefore, total moment $= \frac{8}{8} \times 275{,}000 + \frac{8}{8} \times 251{,}000$
$= 197{,}250$ inch pounds.

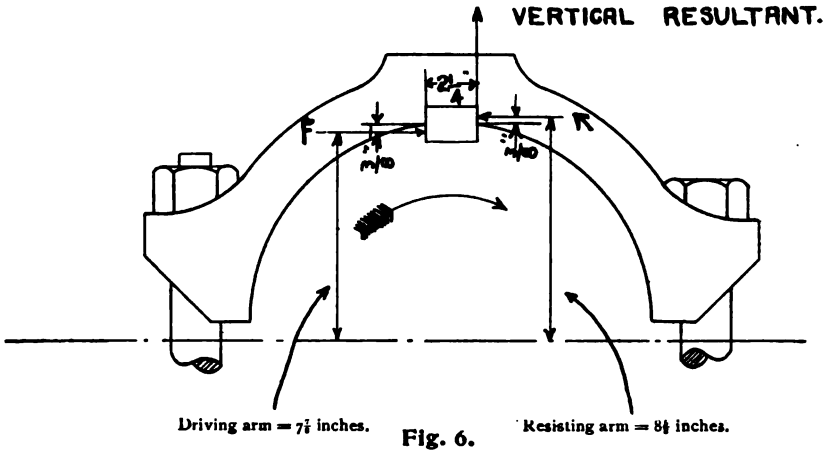

Fig. 6.

This may be considered a moment tending to roll the key in the direction indicated in Fig. 7. It is resisted by the vertical forces p and p_1, which act at an arm equal to width of key $= 2\frac{1}{4}$ inches.

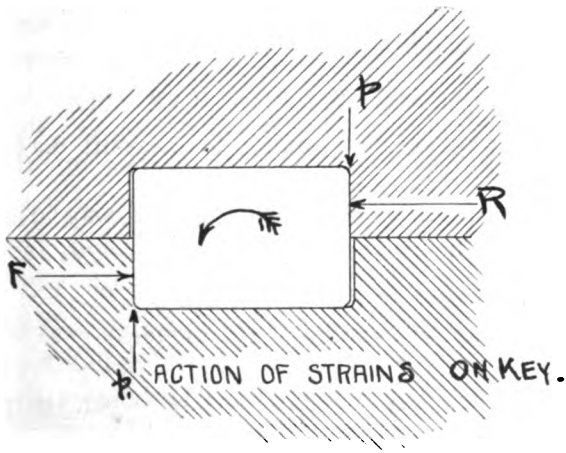

Fig. 7.

Hence the vertical pressure due to the moment tending to roll the key $= 197{,}250 \div 2\frac{1}{4}$ inches $= 87{,}670$ pounds.

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This is, therefore, the force which the bolts have to resist in keeping the halves of the coupling together.

Note.—The driving and resisting forces do not act on planes radial to the axes, and this has a further tendency to spread the coupling, but it is not worth considering in this case, the angle being so small that its sine is practically zero.

STRESS ON BOLTS.

Spreading effect of each key = 87,670 pounds.
Total for both keys on one side = 175,340 pounds.
This is overcome by the 16 bolts of 2 inches diameter.
So the stress on the bolts due to this action of the key =

$$\frac{175,340}{16} = 10,960 \text{ pounds each.}$$

Area of 2-inch bolts at root of thread = 2.302 square inches.
Hence, stress per square inch on bolts =

$$\frac{10,960}{16} = 4,760 \text{ pounds.}$$

This is a safe margin for the bolts, for the keys cannot set up even so much stress as this, owing to the clamping effect of the coupling taking off a part of their work.

OUTBOARD COUPLING, SLEEVE TYPE.

Moment = 4,335,000 inch pounds to transmit 8,250 horse-power.

Mean radius of center of pressure for keyway in shaft = $\frac{1}{2}$ ($8\frac{1}{4}$ inches — $\frac{3}{8}$ inch + $6\frac{7}{8}$ inches — $\frac{3}{8}$ inch) = $7\frac{3}{16}$ inches.

Depth of key = $\frac{3}{4}$ inch.
Length of key = $32\frac{3}{4}$ inches.

$$\text{Pressure on each key} = \frac{4,335,000}{7\frac{3}{16}} \times \frac{1}{2} = 301,350 \text{ pounds.}$$

$$\text{Crushing pressure} = \frac{301,350}{\frac{3}{4} \times 32\frac{3}{4}} = 12,250 \text{ pounds per square inch.}$$

Width of key = $2\frac{1}{4}$ inches = $\frac{3}{4} \times 3$.

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Hence, shearing stress $= \frac{1}{3}$ of crushing pressure (approximately) $= 4{,}080$ pounds per square inch.

Note.—Both these stresses are very moderate, but are sure to be greatly increased by the impossibility of fitting keys inside of long sleeve so that they will receive uniform stress.

Stresses on cotter key driven through shaft and coupling.

Width of key $= 2$ inches.

Depth in contact with sleeve $= 3\frac{1}{4}$ inches at each end.

Hence, area of key under compression =

2×2 inches $\times 3\frac{1}{4}$ inches $= 13$ square inches.

Crushing pressure $= \frac{83{,}560}{13} = 6{,}428$ pounds per square inch.

Shearing area of key $= 2 \times 2$ inches $\times 7$ inches $= 28$ square inches.

Shearing stress $= \frac{83{,}560}{28}$

$= 3{,}000$ pounds per square inch.

Note.—These stresses are in addition to the initial stresses set up by driving the cotter keys into place; this, of course, may be anything, depending on the weight of sledge, etc., used for driving in. When drawing the last shaft into place in the sleeve, the tail shaft, weighing over 15,000 pounds, must be moved at each blow on the cotter. The great inertia of this weight necessitates a great stress on the cotter in order to move it, and leaves a doubt as to whether it is all the way in.

Stress on sleeve at junction of shafts due to torsion.

Moment $= 4{,}335{,}000$ inch pounds $= \frac{\pi S}{16D}(D^4 - d^4)$.

Where $D =$ outside diameter of sleeve $= 21\frac{3}{4}$ inches,

Where $d =$ inside diameter of sleeve $= 13\frac{3}{4}$ inches,

$$S = \frac{4{,}335{,}000 \times 16 \times D}{\pi \times (D^4 - d^4)} = 2{,}450 \text{ pounds per square inch.}$$

This is very moderate, and shows that less metal could be used by making the sleeve less in diameter at its middle, and still have ample strength.

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L. D. LOVEKIN PATENT LOOSE COUPLING AS DESIGNED FOR OUTBOARD-SHAFT COUPLING OF U. S. BATTLESHIP "*KANSAS*." COMPARISON WITH TYPE ADOPTED BY U. S. NAVY.

	U. S. Navy, sleeve type.	Lovekin, clamp type.
Weight of coupling complete, except casing. This weight is the excess over the normal section of shafting of 16½ inches outside diameter, 9½ inches inside diameter, pounds, approximately	3,925	2,515
Difference in types, pounds	1,410	
Two keys on each shaft, dimensions, inches	1½×2¼×32¾	1½×2¼×20
Crushing pressure on keys to resist torsion, pounds per square inch	12,250	†18,320
Shearing force on keys to resist torsion, pounds per square inch	4,080	†5,840
Crushing pressure to resist thrust of propeller, pounds per square inch	*6,428	†‡4,120
Shearing force to resist thrust of propeller, pounds per square inch	*3,000	†‡2,090
Stress in coupling opposite junction of shafts due to torsion, pounds per square inch	2,450	3,210
Stress due to torsion in normal section of shaft, pounds	5,500	5,500
Diameter of hole in shaft at coupling, inches	6	7½

* These are the stresses in cotter key when going astern, in addition to initial stress due to driving in.

† These stresses are materially decreased by the clamping effect of the Lovekin coupling. The corresponding stresses in the sleeve type are especially liable to increase due to the impossibility of fitting accurately.

‡ These are the stresses in clamping ring when going astern.

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DESCRIPTION OF STEAMSHIP *DAKOTA* AND HER OFFICIAL TRIAL PERFORMANCE.

By A. GUNDERSEN, ASSOCIATE.

Steamship *Dakota* is the sister ship of steamship *Minnesota*, designed by Mr. Charles R. Hanscom and built by the Eastern Shipbuilding Company, New London, Connecticut, for the Great Northern Steamship Company's trans-Pacific line between Seattle, Wash., and the Orient.

Steamship *Dakota* is of the type called intermediate, which has of late years become so popular on the Atlantic—that is to say, she is designed neither for passenger service exclusively nor for cargo carrying only, but for combined cargo and passenger traffic in conjunction with the Great Northern Railway Company.

The vessel has accommodation for 206 cabin passengers, 66 intermediate cabin passengers, 735 troops or steerage passengers and a crew of about 250 men.

The first-cabin staterooms are located in the midship house above upper deck and on main deck aft of machinery space. The first-cabin dining saloon is on the upper deck, in the forward end of the midship house, with a seating capacity of 190 passengers. From the dining saloon a grand stairway leads to the promenade-deck hall, at the forward end of which are located the library and ladies' boudoir. Two stairways lead from promenade to the bridge-deck social hall.

The smoking room is located on the bridge deck, forward of the engine hatch casings; extends the full width of the house, and is arranged in a convenient manner. Immediately abaft of the smoking room, between the engine-room hatch casings, are the barroom, toilet rooms and the barber shop. On the promenade and bridge decks several commodious suites are arranged, consisting of bed room, sitting room and bath room.

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The good features in this passenger accommodation are the splendid, wide promenade spaces, the light and airy halls, staterooms and corridors, the facility of getting from one deck to another and accessibility of toilet rooms.

The intermediate-cabin passengers are located on the main deck, forward of hatch No. 5, and the troops or steerage passengers are on the same deck abaft of first-cabin staterooms.

The captain and officers are located in a large house on boat deck, and the chief engineer, assistant engineers, chief electrician and junior engineers have rooms on port side of main deck, abreast of machinery space.

The passenger and crew spaces are ventilated mechanically in addition to the usual natural ventilation. Fresh air is supplied into the passages and the vitiated air is exhausted direct from the rooms. In all large public rooms the exhaust is in the ceiling and the supply comes in at sides. The fresh air supplied to living quarters by the mechanical ventilation is cleansed and heated or cooled by passing over coils for that purpose.

The galleys are on the main deck forward of boiler hatch casing; they are divided by a passage which leads from the first-cabin buffet pantry, adjoining the café on starboard side of main deck, and the officers' pantry, between the main boiler uptakes, to the intermediate-cabin pantry. The first-cabin pantry is on the upper deck, immediately over the galley; communication between the two is by dumb waiter and stairway. At the forward end of the passageway above referred to, and close to the intermediate-cabin pantry, an electric elevator leads to the ship provision storerooms below.

The quartermasters, boatswains, carpenters, water tenders, electric and refrigerating engineers are berthed in the forward deck house on upper deck, which also contains the refrigerating machinery. The laundry men have their quarters in the after deck house on upper deck, in the same house as the laundry machinery is located. The waiters, oilers, seamen and firemen are berthed forward. They are divided into watches and have wash and toilet rooms conveniently near their quarters.

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The hull is constructed of open-hearth steel throughout and built to class 100-A1 Lloyd's awning-deck type.

The double bottom is six feet deep at center, built on the cellular system, extending the entire length of the vessel and divided into twenty-nine watertight compartments. The longitudinal bulkhead, which is 18 inches off center and directly over the starboard vertical keel, extends from end to end of vessel, from tank top to main deck, thus dividing her into two parts. The hull is divided transversely by bulkheads into thirteen watertight compartments. The peak bulkheads are watertight from tank top to upper deck, and the other transverse bulkheads to main deck.

The cargo hatches are in pairs, one for each hold, on each side of the longitudinal bulkhead.

The five lower decks are of steel, continuous the entire length of hull, and the three steel decks in the midship deck house are fitted with expansion joints.

PRINCIPAL DIMENSIONS OF HULL.

Length between perpendiculars, feet and inches	608- 0
over all, feet and inches	627- 9
Beam, extreme, feet and inches	73- 6
Depth of ballast tanks at center, feet and inches	6- 0
Height from tank top to orlop deck at center, feet and inches	9- 6
orlop deck to lower deck, feet and inches	9- 6
lower deck to 'tween deck, feet and inches	9- 6
'tween deck to main deck at side, feet and inches	11- 6
main deck to upper deck, feet and inches	8- 0
upper deck to promenade deck, feet and inches	8- 6
promenade deck to bridge deck, feet and inches	8- 0
bridge deck to boat deck, feet and inches	8- 0
boat deck to captain's bridge, feet and inches	7- 8
upper navigating bridge to bottom of keel, feet and inches	88- 4
Shear forward on upper deck, feet and inches	8- 6
aft on upper deck, feet and inches	6- 6
Block coefficient at a draught of 32 feet 6 inches	0.76
Prismatic coefficient at a draught of 32 feet 6 inches	0.80
Coefficient of fineness of 32 feet 6 inches water line	0.88
Gross tonnage, measured by U. S. custom officials	20,714.32
Net tonnage, measured by U. S. custom officials	13,305.58
Total water ballast, including fresh-water tanks in double bottom, tons	4,000

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Total capacity of fresh-water tanks, tons	1,314
coal bunkers, including reserve bunkers, tons	5,226
deadweight carrying capacity in tons of 2,240 pounds	22,250
Displacement when fully loaded to a draught of 38 feet (Plimsoll mark), tons	37,500

MAIN ENGINES.

There are two main engines, each in a watertight compartment. They are outboard-turning, direct-acting, three cylinder, triple-expansion engines, built by the Midvale Steel Co., Philadelphia, Pa. The arrangement of cylinders and valves is as follows, commencing forward: High-pressure cylinder; high-pressure piston valve; low-pressure cylinder; low-pressure double-ported slide valve; middle-pressure cylinder and the middle-pressure piston valve at the after end.

The cylinders and steam chests are of cast iron, in four separate castings, bolted together. The H.P. cylinder and chest, the M.P. cylinder and chest, the L.P. cylinder and the L.P. valve chest, each being separate castings.

Distance pieces are placed between high and low-pressure cylinders, and between M.P. cylinder and L.P. valve chest, allowance being made for expansion. The H.P. steam chest has an 11-inch diameter opening on column side for connection of steam stop valve, and exhausts into a cast-iron T pipe, 15½ inches in diameter, bolted to H.P. chest at back, thence through a 15½-inch copper pipe with a slip joint at M.P. chest. The M.P. cylinder exhausts through two rectangular passages, each 18 inches by 12½ inches, made a part of cylinder casting, while the L.P. cylinder exhausts through a 32-inch diameter copper pipe, with a slip joint at condenser.

The clothing material for all cylinders and steam chests is composed of 85 per cent. carbonate of magnesia, and 15 per cent. asbestos fiber, lagged with galvanized steel plates with vertical joints.

The framing is of three cast-iron inverted Y housings on the inboard side, with six heavy wrought-steel columns on the outboard side, the housings carrying the guides, reversing engine and brackets for reversing-shaft bearings, and the col-

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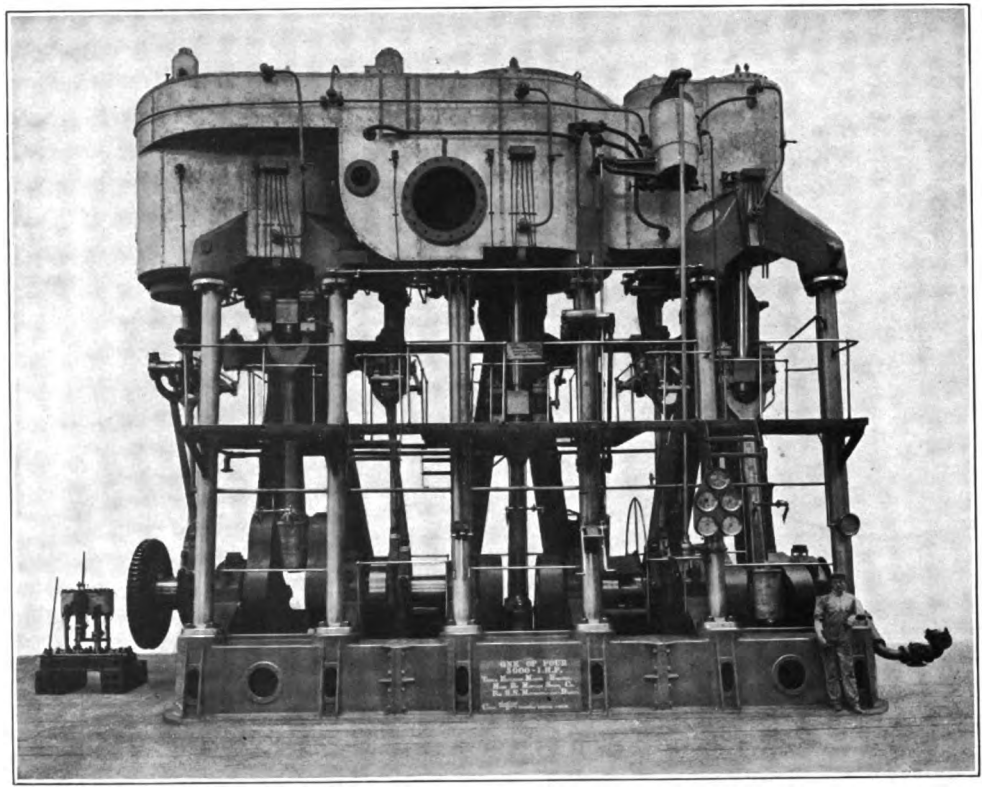

Fig. 1.

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umns carrying the handling and drain gear. The framing is of such proportions as to be thoroughly stiff and rigid, obviating the use of bracing beyond tieing the engines together with athwartship ties at top of housings.

The bed plate is of cast-iron, box section, very deep, and in three sections. The bottom is made to suit the tank top, being firmly bolted to same with one hundred and fifty-eight 1½-inch stud bolts in each bed plate, screwed into tank-top plates and flange of stiffeners with nuts underneath.

There are two main bearings to each crank shaft, and the main bearing caps are of cast steel, lined with Parsons white metal ⅝-inch thick; the bottom boxes are also of cast steel, with composition bushes lined with the same metal of the same thickness as in caps. The bottom boxes are circular, to facilitate removal without lifting of crank shafts.

All the cylinders are jacketed around the working liners. A 1½-inch copper pipe connected to throttle-valve casing supplies steam at working pressure to the high-pressure jackets, and from thence, by Leslie's reducing valves, goes to the M.P. and L.P. jackets, the drop in pressure causing the re-evaporation of the condensed water. This insures all jackets being perfectly drained, and limits the waste due to passage of steam with water through the trap from the one connection at bottom of L.P. jackets.

The cylinder and valve-chest covers are of cast iron, of the box section, and have rolled diamond plates on top. The high and middle-pressure valve-stem guide brackets are cast on the bottom valve-chest covers, and the low-pressure valve-stem guide bracket is bolted to the bottom of low-pressure valve-chest cover. The valve-stem guides are fitted with hard composition boxes, in halves, and held in place by the cap bolts. All the main valves are operated by Stephenson double-bar, link motion with adjustable cut-offs, actuated by screws in slotted arms on the reversing shaft, which may be varied from 0.513 to 0.798 of stroke for H.P., 0.514 to 0.731 for M.P., and 0.526 to 0.731 for L.P. All valves have a common stroke of 10 inches when reversing links are in their normal position. The

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middle and low-pressure valves are fitted with Joy's assistant cylinders, 6¾ inches and 8¾ inches diameter, respectively, while the high-pressure valve is partially balanced by having its upper portion enlarged in diameter. The low-pressure, as before stated, is a double-ported slide valve, and a circular relief frame, 56½ inches outside diameter, is fitted to the L.P. valve-chest on which the back of valve slides. The space inside of this relief ring has a 1-inch pipe connected to the main exhaust pipe.

For further information of valve motion see valve diagram and tables of valve events on Fig. 2.

The body of all main valves is of cast iron; the H.P. and M.P. piston valves have each two solid cast-iron packing rings top and bottom, and the L.P. slide valve works on a cast-iron false face, 1¾ inches thick, bolted to the L.P. cylinder face.

The handling gear, located on the column side of engine (outboard side), is operated from the working platform on the engine-room floor.

On the after column of H.P. cylinder is an 18-inch diameter hand wheel operating the stop valve, which is of the piston balance type, 11 inches in diameter, bolted to the front of H.P. steam chest.

The cock admitting steam to the balance piston is worked by a lever at the same column. At the forward column of the low-pressure cylinder are the levers for throttle valve and reversing engine and the hand wheels for opening and closing the pass-over valves to M.P. and L.P. receivers.

The throttle valve is of the butterfly type, 11 inches in diameter, bolted to the under side of stop valve. This valve is automatically controlled by an Aspinall governor, which is carried on a special lever worked from the low-pressure crosshead. The governor commences to act at seventy-five revolutions per minute, and at ninety revolutions per minute the throttle valve is shut. A disconnecting arrangement is provided for disconnecting the governor gear, enabling the throttle to be worked directly by the hand lever.

On the after column of the low-pressure cylinder are the

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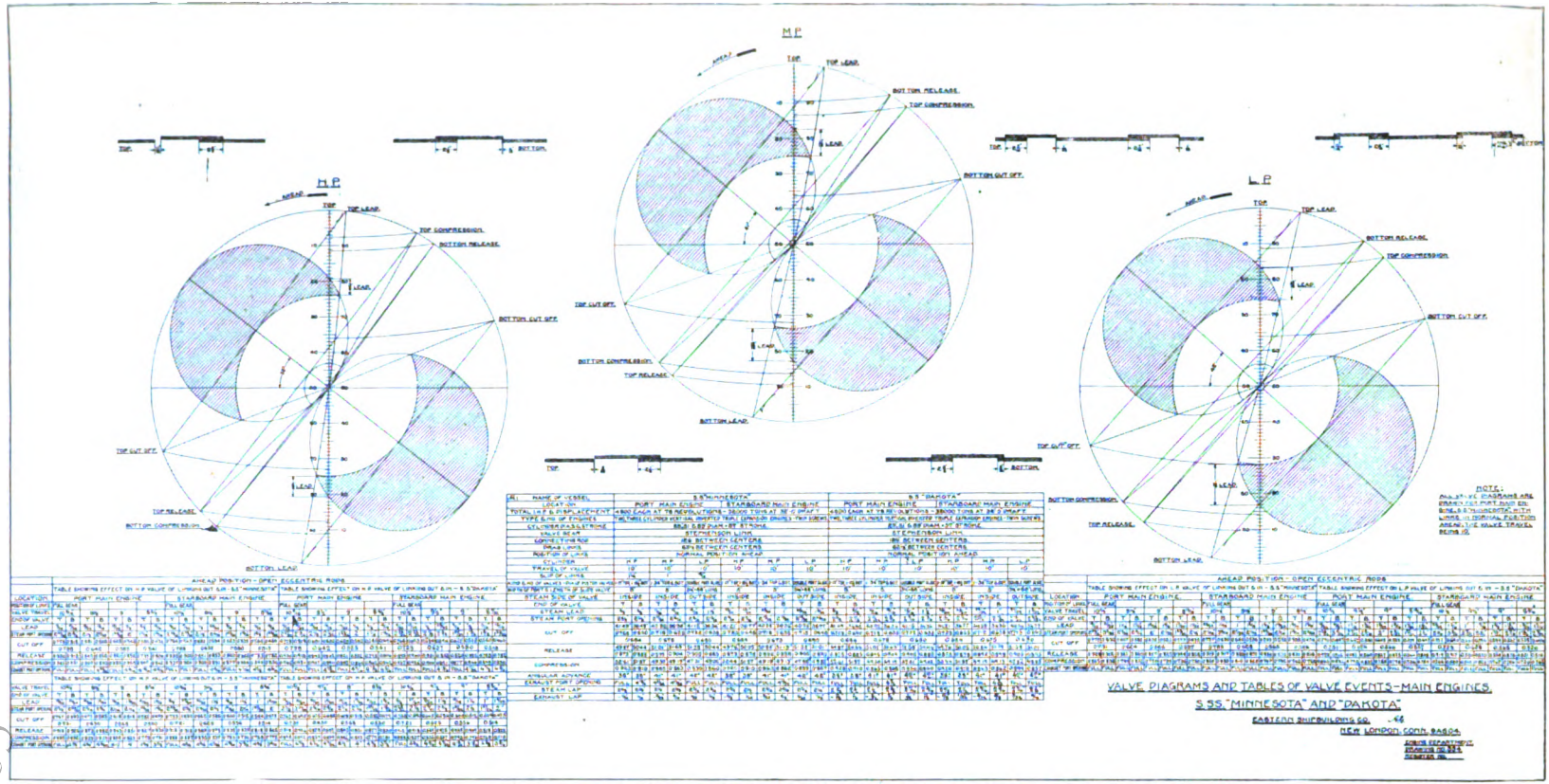

Fig. 2.

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levers for operating the drain cocks, one lever for each cylinder. The cylinder cocks are 2 inches in diameter, and the steam chest cocks are 1¼ inches diameter, connected by separate levers to a 1¼-inch shaft, led fore-and-aft at front of upper column flanges. The operating lever for each cylinder opens or closes the cylinder and chest drains of that cylinder simultaneously. Particular care has been taken to arrange the passages as well as the relief valves so as to keep the cylinders and ports clear of water.

CYLINDER AND VALVE-CHEST DATA.

	H.P.	M.P.	L.P.
Diameter of cylinders, inches	29	51	89
Stroke, inches	57	57	57
Area of cylinders, square inches	660.5	2,042.8	6,221.1
Volume of cylinders, cubic inches	37,648.5	116,439.6	354,602.7
piston rods, cubic inches	2,865	2,865	2,865
Cut-off of stroke, top	0.71	0.712	0.712
bottom	0.64	0.638	0.638
Clearance, top:			
In cylinders, cubic inches	307	1,084	2,627
Ports and passages, cubic inches	5,000	15,500	24,239
Total, cubic inches	5,307	16,584	26,866
Clearance, bottom:			
In cylinders, cubic inches	381	1,578	4,305
Ports and passages, cubic inches	5,148	16,144	26,046
Total, cubic inches	5,529	17,722	30,351
Avge. perctge. clearance, top and bottom,	14.96	14.91	8.1
Volume of cylinders to cut-off, top, cubic inches	26,730	82,905	252,477
Volume of cylinders to cut-off, bottom, cubic inches	22,261	72,461	224,409
Volume of cylinders to cut off, average, cubic inches	24,495.5	77,683	238,443
Volume of cylinders to cut off, average, including clearance, cubic inches	29,913.5	94,836	267,051.5

Ratio of expansion:

$$\text{Nominal ratio} = \frac{\text{Vol. L.P}}{\text{Vol. H.P. to C.O.}} \ldots\ldots 14.48$$

without regard to clearance and release.

$$\text{Ratio} \ldots\ldots = \frac{\text{Vol. L.P. and clearance}}{\text{Vol. H.P. to C.O. and clearance}} \ldots\ldots 12.81$$

not considering release in L.P.

$$\text{Actual ratio} = \frac{\text{Vol. L.P. to release and clearance}}{\text{Vol. H.P. to C.O. and clearance}} \ldots\ldots 11.63$$

release in L.P. cylinder = 0.9 stroke.

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	H.P	M.P.	L.P.
Thickness of cylinder body, inches	$1\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{3}{4}$
liner, inches	$1\frac{5}{8}$	$1\frac{5}{8}$	$1\frac{5}{8}$
jacket space, inches	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$
valve-chest liner, inches	$1\frac{3}{16}$	$1\frac{1}{4}$	...
Number of studs for cylinder cover	31	38	48
Diameter of studs for cylinder cover, inches	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$
stud pitch circle, cylinder cover, inches	38	60	98
Number of studs for valve-chest cover	14	20	34
Diameter studs, valve-chest cover, inches	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$
pitch circle, chest cover, inches	25	$39\frac{3}{4}$	87
Number tap bolts for cylinder liner	26	32	52
Diameter tap bolts for liner, inches	$1\frac{1}{4}$	1	1
pitch bolt circle liner, inches	...	49	87
Number of piston valves	1	1	...
double-ported slide valves	...	...	1
Diameter piston valves, inches { top	17	34	...
Diameter piston valves, inches { bottom	$16\frac{1}{2}$		
Size of slide valves, inches			59×95
Depth of steam port, inches	5	5	(2)$3\frac{1}{4}$
Width of steam port, inches	31	61	88
Diameter cylinder relief valve, top and bottom, inches	3	4	6

PISTONS.

The H.P. piston is of cast iron, flat, and of hollow section; the other pistons are of cast steel, conical in shape. All piston followers, bull rings and packing rings are of cast iron. The packing rings are designed to prevent expansion beyond a given diameter, but allow lineal expansion.

	H.P.	M.P.	L.P.
Depth of piston, over all, inches	10	$23\frac{3}{4}$	$23\frac{3}{4}$
at periphery, inches	10	$8\frac{1}{2}$	$8\frac{1}{2}$
Number of packing rings	2	2	2
Width of packing rings, bearing surface, inches	1	$1\frac{1}{8}$	$\frac{3}{4}$
Thickness of packing rings, inches	$1\frac{1}{4}$	$1\frac{1}{4}$	$\frac{7}{8}$
Number of follower bolts	6	12	20
Diameter of follower bolts, inches	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$

PISTON RODS AND VALVE STEMS.

The piston rods and valve stems are of high-grade carbon steel, finished all over and oil tempered. The ends of piston rods are tapered to fit the pistons and crossheads. The lower end of valve stems is forged large enough for composition bush. The cap and the lower box are cast in one, of the same hard composition as the top half. Piston rods and valve-stem

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stuffing boxes are fitted with Katzenstein's metallic packing, and have a small stuffing box for soft packing on the outside of the metallic packing.

Diameter of piston rods, inches	8
threads, top and bottom, inches	6
Length piston rods, over all, inches	129
Diameter valve stems through guide and gland, inches	$5\frac{1}{2}$
valve, inches	4
Number of bolts in valve-stem brasses	2
Diameter bolts in valve-stem brasses, inches	$2\frac{1}{4}$

MAIN VALVE LINKS AND LINK BLOCKS.

The links and link blocks are of high-grade carbon steel, finished all over and oil tempered.

The link blocks have at each end a pair of jaws to span the corresponding bar of the link. The jaws are fitted with composition gibs, finished to the curve of the links, and have tap bolts for adjustment. The pins for suspension rods are in the center of links.

Depth of link bars, inches	$6\frac{3}{4}$
Width of link bars, inches	$2\frac{1}{4}$
Distance between bars, inches	$8\frac{1}{2}$
pins for eccentric rods, inches	30
Diameter of pins for eccentric rods, inches	4
Length of pins for eccentric rods, inches	$4\frac{1}{16}$
Diameter of suspension-rod pins, inches	$2\frac{3}{4}$
Length of suspension-rod pins, inches	$2\frac{3}{4}$
Diameter of link-block pin, inches	7
axial hole through link-block pin, inches	3
Distance between link-block jaws, inches	6

CONNECTING RODS, ECCENTRIC RODS AND ECCENTRICS.

The above mentioned rods are of open-hearth wrought steel, annealed after forging, and finished all over. The upper ends of connecting rods are forked for crosshead brasses; these brasses are of phosphor-bronze, and the crank-pin brasses are of composition lined with Parsons white metal, $\frac{5}{8}$ inch thick. The eccentric rods are also forked to span the links, and fitted with adjustable brasses. The eccentric straps are of cast steel lined with white metal. The eccentrics are made in two parts; the larger half is of cast iron, and the smaller of cast steel.

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Length of connecting rods, center to center, inches	126
Diameter of rod at crank end, inches	10
fork end, inches	8
Number of bolts in crank-pin brasses	2
crosshead brasses	4
Diameter of bolts in crank-pin brasses, inches	$4\frac{3}{16}$
crosshead brasses, inches	$3\frac{3}{16}$
Length of eccentric rods, center to center, inches	117
Diameter of eccentric rods at top, inches	$3\frac{3}{4}$
bottom, inches	$4\frac{3}{4}$
Number of bolts in top brasses, each rod	4
Diameter of bolts in top brasses, inches	$1\frac{3}{4}$
Width of eccentric straps, inches	6
Diameter of bolts in eccentric straps, inches	$2\frac{3}{16}$
studs in top of eccentric straps, inches	$2\frac{1}{2}$
eccentrics, inches	$32\frac{1}{2}$
bolts in eccentrics, inches	$2\frac{1}{2}$

CROSSHEAD GUIDES, CROSSHEADS AND CROSSHEAD SLIPPERS.

The crosshead guides are of the slipper type, of cast iron and bolted to housings, and are hollow for water circulation. The crosshead slippers are of cast steel, bolted to crossheads and filled with white metal. The crossheads are of wrought steel.

Diameter of crosshead pins, inches	10
Length of crosshead pins, each, inches	11
slipper, inches	30
Width of crosshead slipper, inches	24
Length of backing surface, inches	30
Width of backing surface, inches	16

CRANK SHAFTS.

The crank shafts are of the built-up type, each in three duplicate and interchangeable sections, and the sequence of cranks is as follows: H.P., M.P. and L.P.

Length of each section, inches	$125\frac{1}{2}$
Diameter of shafts and pin, inches	18
axial hole through shafts and pin, inches	8
Distance between crank webs, inches	20
Diameter of coupling flanges, inches	$31\frac{1}{2}$
Width of coupling flanges, inches	$4\frac{1}{2}$
Number of coupling bolts	9
Diameter of coupling bolts, inches	*$3-3\frac{3}{4}$

* Taper in 9 inches.

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Diameter of crank webs, at ends, inches	37½
Thickness of crank webs, inches	12½
Number of main bearings	6
Length of main bearings, inches	18
Number of main bearing bolts per bearing	2
Diameter of main bearing bolts, inches	3 7/16

SHAFTING AND THRUST BEARING.

All shafts, except crank webs, are of high-grade carbon steel, oil tempered; tensile strength, 75,000 to 85,000 pounds; elastic limit, not less than 45,000 pounds; elongation, 24 per cent. in 2 inches.

Crank webs, tensile strength, 58,000 to 65,000 pounds; elastic limit, not less than 35,000 pounds; elongation, 25 per cent. in 2 inches; annealed after forging.

The shafting is installed in the following order: Aft of each crank shaft, thrust shaft, forward Verity coupling shaft, six lengths of line shafts, after Verity coupling shaft and the propeller shaft. The propeller shafts are encased in composition casings the whole length of stern tube.

The thrust bearing is of the horseshoe type. The body is of cast iron, so shaped that it forms a reservoir for oil with a bearing at each end.

The thrust shoes are of cast steel, hollow for water circulation, and faced with white metal.

DATA FOR SHAFTING.

Diameter of thrust shaft, inches	18
thrust-shaft hole, inches	8
Number of thrust-shaft collars	9
Outside diameter of thrust-shaft collar, inches	27⅝
Width of thrust-shaft collars, inches	2½
Space between thrust-shaft collars, inches	5½
Length of thrust-shaft, feet and inches	14- 5½
bearings, each, inches	15¾
Number of thrust shoes	9
Diameter of thrust-bearing side rods, inches	3¾
Verity coupling shafts, inches	17
shaft hole	7¾
Length of Verity coupling shafts, feet and inches	7- 7½
Diameter of line shafts, inches	17
Lengths of line shafts, feet and inches	22- 6

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Length of spring bearings, inches	18
Number of spring bearings in each tunnel, aft of thrust recess	14
Diameter of propeller shafts, inches	19
shaft hole, inches	8¼
Length of propeller shafts, feet and inches	36–10¾
Thickness of casing at bearings, inches	1⅛
Length of stern bush bearing, inches	80
bearing in stern-tube stuffing box, inches	50

All shaft couplings forming part of Verity coupling, of which there are two on each line of shafting, one at after end of thrust shaft and one at forward end of propeller shaft, are 34 inches in diameter. All other shaft couplings are the same as on crank shaft.

REVERSING ENGINES.

The reversing engine is located forward of L.P. housing, with its steam cylinder bolted to same. There is an oil-control cylinder connected to the steam cylinder by tie rods, which serve as crosshead guides. One piston rod extends through each cylinder and is made of forged steel. There is also provided an oil pump, for hand reversing, of the single-acting plunger type.

The reverse shaft is near the top of the inboard side of the housings and the levers are connected to the crosshead of the reverse engine by two forged-steel connecting rods. Differential levers control the valve mechanism.

DATA FOR ENGINE.

Diameter of steam cylinder, inches	14
oil cylinder, inches	7
steam and oil piston rods, inches	3¼
tail rod, inches	3¼
Stroke, inches	27
Length of crosshead bearing, inches	7
Diameter of crosshead pins, inches	3½
Length of crosshead pins, each, inches	3½
Diameter of tie rods between cylinders, inches	2⅞
forming crosshead guides, inches	3
threads at ends, inches	2¼
connecting rods, ends, inches	3
middle, inches	3½
Length of connecting rods, inches	82
lever on reverse shaft, inches	27

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Diameter of pins in reverse lever, inches	3½
Length of reverse-shaft levers, inches	27
suspension rods to links, inches	63½
Diameter of reverse shaft, inches	8
at bearings, inches	9
Length of reverse shaft bearings, inches	10

TURNING ENGINE.

The turning engine is a double-cylinder vertical engine, operated by a change valve and located in port engine room close to bulkhead on frame No. 82. At each end of the crank is a clutch fitting into corresponding clutches on athwartship shafts, extending over to each of the main engines. These clutches are so arranged and operated by hand levers that only one main engine can be turned at a time. This athwartship shaft drives a worm and wheel (ratio, 30:1) at each main engine, this latter wheel being keyed on a vertical shaft with worm on upper end, which worm meshes into the turning wheel on after crank-shaft coupling of main engines. (Ratio of last worm and wheel, 60:1.)

DATA OF TURNING GEAR.

Number of steam cylinders	2
Diameter of steam cylinders, inches	6
piston rods, inches	1 9/16
valves, inches	3
change valves, inches	3
Stroke, inches	6
Valve travel, inches	1¼
Crank angle, degrees	90
Diameter of crank shaft, inches	2½
Length of crank-shaft bearings, inches	2¾
Diameter pitch circle of horizontal worm, inches	3.66
wheel, inches	16.71
Number of teeth in wheel	30
Diameter pitch circle of vertical worm, inches	9
wheel, inches	57.3
Number of teeth	60
Diameter of vertical shaft at worm, inches	4
bearing, inches	5½
Depth of bearing in vertical bracket, inches	19

WATER AND OIL SERVICE.

The water-service pipe is connected to a nozzle on the main circulating-pump discharge to main condensers. This pipe

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leads below the floor plates to the outboard side of housings, with connections to all main bearings and guides, as well as sprays for external cooling of main bearings and crank pins. There is an extension from the water-service pipe to the thrust bearing, spring bearings and stern-tube stuffing boxes.

The oiling is by wick feed from brass oil boxes on cylinders for main moving parts; separate boxes being fitted for each main bearing and eccentrics, while eccentric-rod pins, links, etc., have separate boxes of locomotive type.

PROPELLERS.

The propeller hubs are of cast steel, fitted on the tapered end of the propeller shaft, and held in place by one longitudinal key and a nut on end of shaft, threaded in the opposite direction to the rotation of the propeller.

The blades are of maganese-bronze, and are fitted into recesses in the hub, and are held in place by steel stud bolts with composition box nuts.

The starboard propeller is right-handed and the port propeller is left-handed.

DATA FOR ONE PROPELLER.

Number of blades	3
Pitch on official speed trial, true screw, feet	22
maximum, feet	23
minimum, feet	21
Helicoidal area, square feet	97
Projected area, square feet	82
Number of stud bolts for blades	27
Diameter of stud bolts, inches	3¼
Width of longitudinal key, inches	4½
Thickness of longitudinal key, inches	2¼
Diameter of nut threads on shaft, inches	13
Length of nut threads on shaft, inches	12
Number of threads per inch	2½

MAIN CONDENSER.

One main condenser is located on the outboard side of each engine room. The shell plating is of steel and the water-heads of cast-iron, the after chest being the one for the entrance and exit of circulating water, entering at bottom and leaving

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at top. Each head has four manholes, 18 inches in diameter, fitted with cast-iron covers. The condenser tube sheets are of rolled composition, with smoothly-finished holes for the tubes, tapped and fitted with composition screw glands for packing the tubes.

DATA FOR ONE CONDENSER.

Outside diameter of shell, feet and inches	6– 8⅞
Thickness of shell, inch	7/16
Length of shell, feet and inches	12– 0
Thickness of composition tube sheets, inches	1⅛
supporting plate, inch	⅜
Number of tubes	2,456
Diameter of tubes, inch	⅞
Thickness of tubes, B.W.G	17
Length of tubes, feet and inches	12– 2
Pitch of tubes, inches	1 5/16
Cooling surface, square feet	6,750

AUXILIARY CONDENSER.

There is one auxiliary condenser in each engine room, located on the outboard side of the hot well, and mounted on a Blake horizontal combined air and circulating pumps of the well-known Wheeler type. The air pump discharges into the filter-tank on top of hotwell; the circulating pump draws from the sea, and discharges through condenser to the overboard delivery.

DATA AUXILIARY CONDENSER.

Cooling surface of one condenser, square feet	850
Diameter of steam cylinder, inches	12
air cylinder, inches	14
water cylinder, inches	14
Stroke, inches	12

MAIN CIRCULATING PUMP.

There is one centrifugal circulating pump in each engine room, driven by a two-cylinder compound vertical inverted engine, located at the after end of main condenser. Each pump draws from the sea and the engine-room bilge, and discharges through the condenser to the overboard delivery. The casing and impeller are of composition.

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DATA FOR PUMP.

Diameter of suction nozzle, inches	20
discharge nozzle, inches	20
impeller, inches	42
Width at tip of blade, inches	3½
Diameter of H.P. cylinder, inches	10
L.P. cylinder, inches	20
piston rod, inches	2¼
Stroke, inches	10
Length of connecting rod, inches	25
Valve travel, inches	2½
Diameter of H.P. piston valve, inches	5¼
L.P. piston valve, inches	7½
crosshead pin, inches	3
Length of crosshead pin, inches	4
Diameter of crank pin, inches	4½
Length of crank pin, inches	6½
Diameter of crank shaft, inches	4½
Crank angle, degrees	90
Rate of discharge, in gallons per minute, at 160 revolvtions	10,000

MAIN AIR PUMP.

There is one Blake air pump in each engine room, located on the inboard side of the main condenser, of the vertical simplex cross-connected featherweight pattern, arranged to run both sides together or either side alone. The air cylinders are arranged with discharge valves at each end, but with no suction or bucket valves at all. Around the middle of each cylinder is an annular port connecting with the suction pipe. The passage of the air piston over this port performs the function of a suction valve.

The air pumps discharge into the filter above hotwell. When cleaning the filter, the connections are such that the air pump can discharge direct into the hotwell.

DATA FOR AIR PUMP.

Number of steam cylinders	2
Diameter of steam cylinders, inches	10
Number of pump cylinders	2
Diameter of pump cylinders, inches	20
Stroke, inches	16
Diameter of suction pipe, inches	12
discharge pipe, inches	10

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HOTWELL AND FILTER TANK.

There is one hotwell in each engine room, located forward of the main air pump. The top of the hotwell is arranged as a filter; the filtering material on speed trial was hay, but was changed to "loofa," in New York, before leaving for the Pacific coast. The filtering chamber is divided by vertical partitions so that the water flows from the bottom of one to the top of the succeeding one. Each compartment in the filter tank is provided with a vapor pipe.

MAIN FEED LIFT PUMP.

There is one Blake lift pump in each engine room, located on the working platform forward of hotwell, of the vertical simplex piston type, with suctions from hotwell and reserve-feed tank and discharges into the jet feed-water heater located on the outboard engine-room casing, between lower and 'tween decks. This pump is operated automatically by a float in hotwell connected to a Fitts chronometer governor valve. Before the feed water reaches the heater it goes through a Rankin grease extractor, in which Turkish toweling is used for filtering material, drawn over perforated brass tubes.

Diameter of steam cylinder, inches	8
water cylinder, inches	12
Stroke, inches	12

MAIN FEED PUMP.

There is one Blake main feed pump in each engine room of the vertical simplex piston type, located on the working platform at side of vessel near forward end of engine room, with suctions from jet heater, hotwell and reserve-feed tank, and discharges into the main feed line, which is 5 inches in diameter. Before the feed water reaches the boilers from this pump it passes through another grease extractor of the Railton & Campbell pattern, in which Turkish toweling is also used for filtering material, in one piece, drawn over a large corrugated brass tube in each filter. This pump is operated automatically by a governor valve connected to a float in the jet

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heater receiver; it can also be operated automatically from the float in the hotwell.

Diameter of steam cylinder, inches	18
water cylinder, inches	10
Stroke, inches	18
Rate of discharge, in gallons per minute, at 32 double strokes, against a water pressure of 325 pounds per square inch and a steam pressure of 125 pounds per square inch	400

AUXILIARY-FEED PUMP.

There is one Blake auxiliary feed pump in each after fire room, located at the side of the vessel, of the vertical compound duplex piston type, with suctions from jet heater, hot well, reserve-feed tank in double bottom under engine room, and the sea, and discharges into main and auxiliary-feed lines to all main boilers, and can also be used for operating the ash ejectors in fire rooms.

Diameter of H.P. steam cylinders, inches	12
L.P. steam cylinders, inches	18½
double-acting water cylinder, inches	8

GENERAL DONKEY PUMP.

There is one general donkey pump on the vessel, located in the port engine room on the orlop deck platform, close to bulkhead on frame No. 82, on the outboard side of the main engine. This pump is also of the Blake make, of the horizontal compound duplex type, with suctions from sea, all ballast tanks and bilge distribution valve boxes, with special connections to forward and after peak tanks, reserve-feed tank and port engine-room bilge, and discharges overboard, through port main condenser, into deck fire and service pipes, into main-feed line on port side, and into reserve-feed tank for filling purposes.

Diameter of high-pressure cylinders, inches	14
low-pressure cylinders, inches	20
double-acting water cylinder, inches	10
Stroke, inches	12

FIRE PUMP.

There is one fire pump on the vessel, located in the starboard engine room, on orlop-deck platform, close to bulkhead

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on frame No. 82. This pump is of the Blake horizontal compound duplex piston type, with suctions from sea, ballast and bilge distributing valve boxes, with independent suction from starboard engine-room bilge, and discharges overboard, through starboard main condenser and into deck fire and service pipes.

Diameter H.P. cylinders, inches	14
L.P. cylinders, inches	20
of double-acting water cylinders, inches	10
Stroke, inches	12
Rate of discharge, in gallons per minute, at 30 double strokes, against a water pressure of 100 pounds per square inch and a steam pressure of 115 pounds per square inch	1,450

BALLAST PUMP.

There is in each engine room, under main condenser, a ballast pump of the Blake horizontal compound duplex piston type, with suctions from sea, all ballast and bilge distributing valve boxes, with independent suction from engine-room bilge, and discharges overboard and through main condenser. Pump on port side also discharges into the deck, fire and service pipes.

Diameter of H.P. cylinders, inches	8
L.P. cylinders, inches	12
of double-acting water cylinders, inches	14
Stroke, inches	18

BOILER TEST PUMP.

In the forward end of port engine room, on the working platform, close to longitudinal bulkhead, is located a Blake horizontal duplex piston pump with suctions from reserve-feed tank and all fresh-water tanks, and discharges into cross pipe connecting the two main-feed lines, and into reserve-feed tank for filling purposes. This pump is also used as a feed pump, and is large enough for feeding two main boilers, when used as donkey boilers, with a steam pressure of 200 pounds per square inch.

Diameter of steam cylinders, inches	6
water cylinders, inches	2¼
Stroke, inches	6
Rate of discharge, in gallons per minute, at 53 double strokes with 140 pounds of steam against a water pressure of 500 pounds per square inch	9

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SANITARY SYSTEM.

The sanitary arrangement is on the pressure system; each of the sanitary pumps has an automatic governor valve, regulating the water pressure throughout the vessel. There is a hot, as well as a cold, water service, both for fresh and salt water, with separate pumps and distributing pipes. The sanitary pumps are of the Blake make, and are located in the port engine room, on orlop-deck platform alongside of longitudinal bulkhead.

The salt-water sanitary pump is a horizontal compound duplex pump, with suction from sea, and discharges into the salt-water system.

Diameter of H.P. steam cylinders, inches	8
L.P. steam cylinders, inches	12
double-acting water cylinder, inches	10¼
Stroke, inches	10
Rate of discharge, in gallons per minute, at 78 double strokes, with a steam pressure of 140 pounds per square inch against a water pressure of 60 pounds per square inch	1,140

The fresh-water sanitary pump is a vertical compound duplex piston pump, with suction from reserve feed tank, all other fresh-water tanks and the aerating tank, and discharges into the fresh-water system.

Diameter of H.P. steam cylinders, inches	5¼
L.P. steam cylinders, inches	7¼
double-acting water cylinder, inches	7¼
Stroke, inches	6
Rate of discharge, in gallons per minute, at 45 double strokes with a steam pressure of 130 pounds per square inch against a water pressure of 45 pounds per square inch	390

The auxiliary sanitary pump is a horizontal duplex piston pump, with suctions from sea, reserve-feed tank, all other fresh-water tanks and the aerating tank, and discharges into both fresh and salt-water systems.

Diameter of steam cylinders, inches	10
water cylinders, inches	10¼
Stroke, inches	10

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The fresh-water lift pump is a vertical simplex piston pump, with suctions from distiller air pump discharge in reserve feed tank and the other fresh-water tanks, and discharges through filters into aerating tank.

Diameter of steam cylinder, inches	4½
water cylinder, inches	6
Stroke, inches	6
Rate of discharge, in gallons per minute, at 100 double strokes, with 125 pounds of steam against a water pressure of 5 pounds per square inch	150

The fresh-water filters, aerating tank and air compressor are located on lower-deck platform, above sanitary pump. Charcoal is used as filtering material in these filters.

The air compressor is of the Blake single-flywheel type, with suction leading from boat deck, and discharges into the aerating tank through a number of holes in perforated pipes at bottom of same, the distilled water entering the aerating tank at top in form of a spray.

Diameter of steam cylinder, inches	6
air cylinder, inches	6
Stroke, inches	7

The discharge from this air compressor also leads into engine rooms, machine shop and fire rooms, with hose connections at different places for use in operating pneumatic tools, and for cleaning boiler and heater tubes, if it should be required.

EVAPORATING PLANT.

The evaporating plant is arranged with four horizontal evaporators, two distillers mounted on a combined air and circulating pump, two vertical feed-water heaters and two horizontal combined feed and brine pumps.

All of the above-mentioned machinery is located in the forward end of the engine rooms, under the orlop-deck platform, with two evaporators in each engine room.

Each evaporator receives steam from the auxiliary steam line, at a pressure of 150 pounds per square inch. The passage

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of the steam is from the front to the back header through all the tubes except the bottom row, through which it is returned to the front header. Steam gague, relief valve and drain connection are made to the front header. A trap, which discharges to the reserve feed tank below, is provided to drain the tubes in each evaporator. The connections are so arranged that the evaporators can be operated single, double, triple and quadruple effect.

DATA FOR EVAPORATORS.

Diameter of shell, inches	63
Length of shell, inches	70
Thickness of shell, inch	$\frac{5}{8}$
heads, inch	$\frac{1}{2}$
Number of units to each evaporator	2
tubes in each unit	35
Diameter of brass tubes, inches	2
Thickness of brass tubes, B.W.G	10
Heating surface (tubes only) in each evaporator, square feet	225
tube sheet and header, each evaporator, square feet	13
total, each evaporator, square feet	238
Capacity of evaporators based on a four-hour test:	
Gallons of water evaporated in 24 hours, the four evaporators working single effect	52,524
When this test commenced the evaporators were perfectly clean.	
Gallons of water evaporated in 24 hours, the four evaporators working double effect	28,170
When this test commenced the evaporators had been in use four hours without being cleaned.	
Gallons of water evaporated in 24 hours, three of the evaporators working triple effect	16,824
When this test commenced the evaporators had been in use for eight hours without being cleaned.	

DISTILLERS.

The distillers are of the Wheeler Admiralty type, each mounted on a combined air and circulating pump, located on the outboard side of evaporators, one in each engine room. The bodies are of cast iron and the cooling surface is obtained from brass tubes, tinned inside and out, expanded into brass tube sheets.

Cooling surface (tubes only) in one condenser, square feet	50

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COMBINED AIR AND CIRCULATING PUMPS.

The air pump discharges into fresh-water lift-pump suction box in the reserve-feed tank and into center tank No. 6 in double bottom. The circulating pump takes water from sea and discharges through distiller overboard.

Diameter of pump steam cylinder, inches	6
air cylinder, inches	8
of water cylinder, inches	8
Stroke, inches	7

EVAPORATOR FEED AND BRINE PUMPS.

The evaporator feed and brine pumps are of the Blake horizontal combined simplex type. The feed pump, with suction from distiller circulating pump discharge, discharges through the vertical feed-water heater into evaporator shells. The brine pump, with suction from bottom of evaporator shells, discharges overboard.

Heating surface of evaporator feed-water heater, square feet	50
Diameter of steam cylinder, inches	4½
feed-pump cylinder, inches	3½
brine-pump cylinder, inches	3½
Stroke, inches	6

MAIN BOILERS.

There are sixteen Niclausse water-tube boilers, designed and built by the Stirling Company, for a maximum coal consumption of 40 pounds per square foot of grate surface per hour, containing 272 elements, constructed for a steam pressure of 250 pounds and arranged fore-and-aft as shown on plate No. 1.

Each boiler is fitted with a horizontal air-heating tube box on top of casing and at back of drum to heat the air before entering the ash pit in connection with the Ellis and Eaves' system of induced draft.

When the fires are worked under the induced-draft system, the air necessary for combustion enters one section of the heater tubes at one end of box and enters the other section at end of box diagonally opposite; that is, that the air travels through one section of heater tubes in one direction, while in

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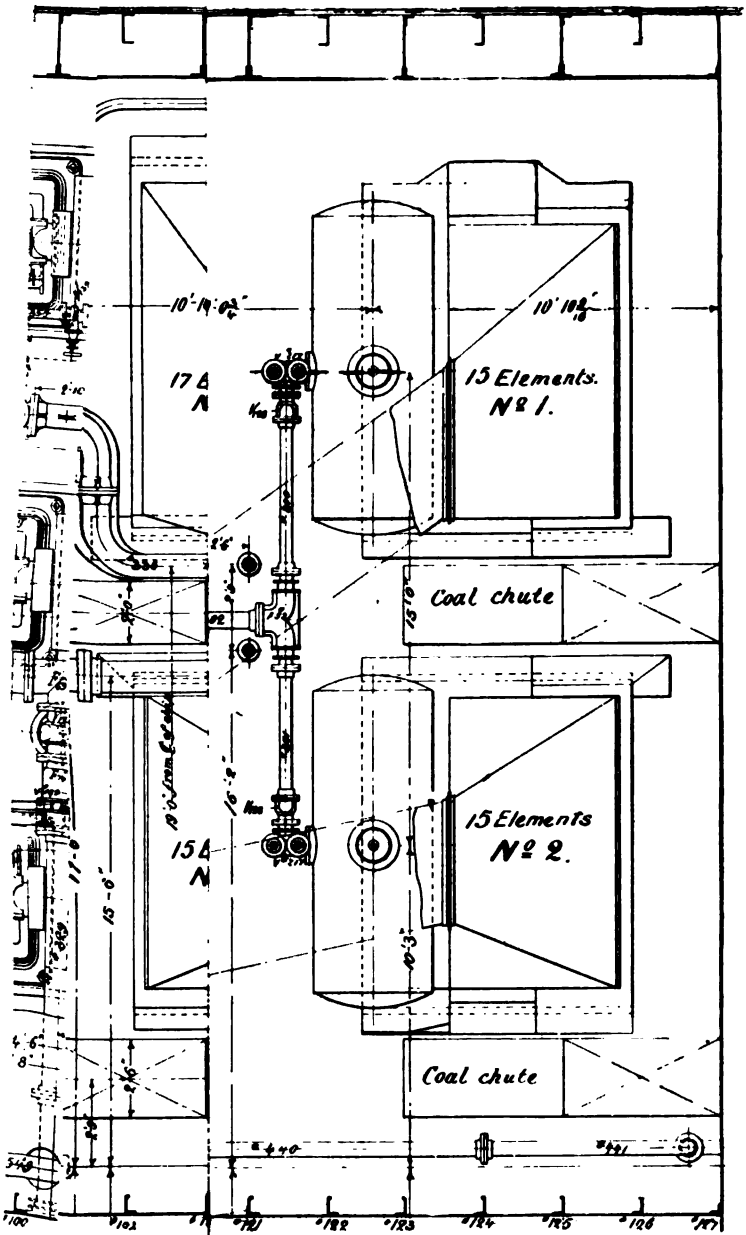

Plate

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the other section of tubes it travels in the opposite direction, thence down at both sides of boiler casing through passages into ash pits. These passageways, forming part of the boiler side casing, are insulated and gradually widen out to the full length of ash pit at bottom. (See Fig. 3.)

DATA FOR BOILERS.

Number of boilers, including all sizes....	16			
	15 element.	17 element.	18 element.	19 element.
Number of boilers in reference to number of elements	6	2	4	4
Number of tubes in each element	24	24	24	24
Diameter of steam drum inside, inches	45	45	45	45
Length steam drum, feet and inches	9- 7 3/4	10-10 1/2	11- 5 1/2	12- 3/4
Thickness of drum plate, inches	15/16	15/16	15/16	15/16
Diameter steam dome, inches	11 29/32	11 29/32	11 29/32	11 29/32
Height steam dome, inches	16	16	16	16
Size rectangular section of headers, ins.	6 37/64 × 7 5/32 × 7/16 inch thick.			
Diameter evaporating tubes, inches	3 1/4	3 1/4	3 1/4	3 1/4
Length evaporating tubes, feet and ins.	7- 4 1/2	7- 4 1/2	7- 4 1/2	7- 4 1/2
Thickness evaporating tubes in the eight lower rows, B.W.G.	6	6	6	6
Thickness evaporating tubes above the eighth row, B.W.G.	8	8	8	8
Diameter of circulating tubes, inches	1 19/32	1 19/32	1 19/32	1 19/32
Length of circulating tubes, ft. and in.	7- 3 13/16	7- 3 13/16	7- 3 13/16	7- 3 13/16
Thickness of circulating tubes, B.W.G.	21	21	21	21
Total heating surface, square feet	13,438	5,077	10,751	11,348
Total heating surface in all boilers, sq. ft.	40,614			
Grate surface in boilers for hand firing, square feet	298	67.5	143	301
Grate surface in boilers with stokers, square feet	59	67.5	152	
Total grate surface, square feet	357	135	295	301
Total grate surface, all boilers, sq. feet	1,088			
Ratio of H.S. to G.S.	37.3 : 1			
Diameter tubes in heater boxes, inches	3	3	3	3
Thickness tubes in heater boxes, B.W.G.	12	12	12	12
Length tubes in heater boxes, ft. and in.	3-10 3/4	4- 6	4- 9 3/8	5- 1 1/4
Total number tubes in heater boxes	2,160	776	1,680	1,680
Sectional area of heater tubes for one boiler, square feet	7.597	8.186	8.863	8.863
Sectional area of space between heater tubes for escape of waste gases in each boiler, square feet	10.83	12.54	13.4	14.44
Sectional area of uptake for each boiler	10.5	11.5	12.5	13

Total sectional area of all branch uptakes, square feet..... 188
Sectional area of the four main uptakes, square feet..... 182
Sectional area of rectangular stack above upper deck, square feet..... 224
The smokestack is double and elliptical in shape.
Outer stack, 19 feet fore-and-aft and 16 feet 3 inches athwartship.
Inner stack, 16 feet fore-and-aft and 13 feet 6 inches athwartship.
Height of stack above base, feet and inches..... .....134- 6
grate, feet and inches.....124-10
boat deck, feet and inches..... 54- 6

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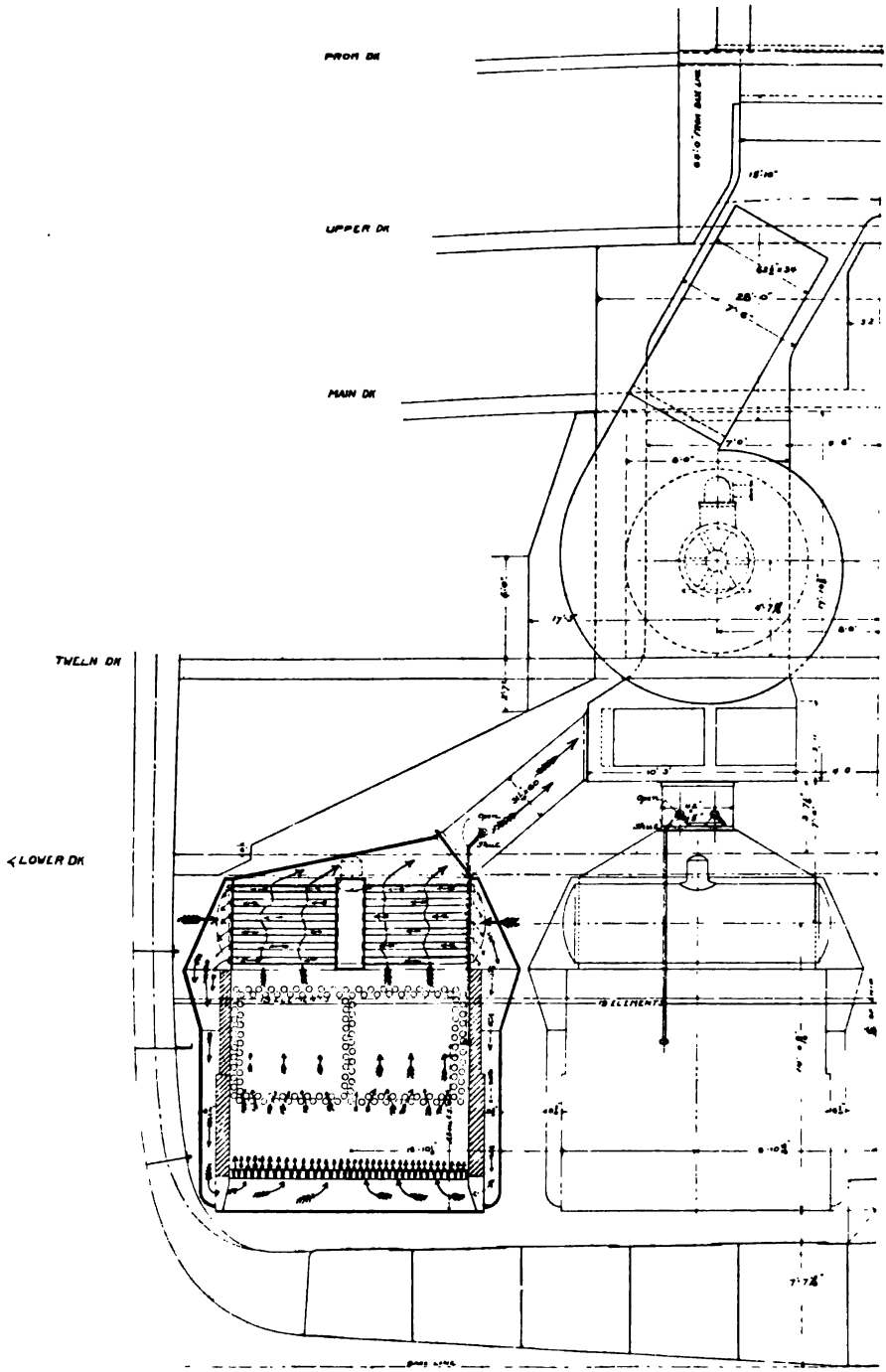

Fig. 3.

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It will be understood that the air is heated while passing through the horizontal tubes on top of boiler by the gases from the fire below, passing on the outside of said tubes before entering into the uptake.

Provision is made on each boiler for letting in and regulating the air supply, from the side passages to top of grate at front and back of fires.

The air passages at side of boilers are fitted with dampers which are operated from the front of boilers. The main uptake on each side of the vessel is divided by a center partition up to the main deck, in which are fitted two large dampers for changing the condition of draft from natural to induced, or *vice-versa.* Each branch uptake is also fitted with a damper which is operated from the fire-room floor.

The four after starboard boilers are fitted with Duluth mechanical stokers, driven by two 7-inch × 6-inch vertical inverted engines, located on the starboard side of longitudinal bulkhead in about the middle of after fire room on level with the orlop-deck grating.

INDUCED-DRAFT FANS.

There are four induced-draft fans, two on each side, located forward and aft of the main uptakes, between main and 'tween-deck gratings. These fans are manufactured by the New York Blower Co.: each of them is direct connected to a Westinghouse automatic compound engine, located in recesses forward and aft of boiler-hatch casings, as will be seen on Fig. 3.

Diameter of fans, inches	108
Width at tip of blades, inches	30
Area of inlet, square feet	30
outlet, square feet	19.25
Diameter of H.P. cylinder, inches	9
L.P. cylinder, inches	15
Stroke, inches	9

STEAM AND EXHAUST PIPES.

The arrangement of steam and exhaust pipes is about the same on both sides, with the exception of steam and exhaust

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pipes to sanitary pumps, which are in the port engine room. The port and starboard main steam lines and the auxiliary steam and exhaust lines are connected in the engine rooms by cross-pipes. There is on each auxiliary steam line an 8-inch Leslie reducing valve, located at forward end of engine room, reducing the boiler pressure to 150 pounds per square inch. On the branch castings on auxiliary steam lines in the after fire rooms are four reducing valves, connected for reducing the boiler pressure to 150 pounds per square inch for steam pipes leading to windlass engines, induced-draft fan engines, auxiliary feed pumps, chain-grate engines, and ash-hoist engines.

From the above mentioned arrangement of reducing valves, it will be understood that all of the auxiliary machinery is supposed to work with a steam pressure of 150 pounds per square inch, but all of them are so constructed that they will stand the full boiler pressure in case the reducing valves should fail to do their duties.

The main steam pipe and auxiliary steam pipe are the same for the eight forward boilers; but, for the eight after boilers these pipes are separate and parallel.

The exhaust steam from all auxiliaries goes through two Cochrane vacuum oil separators before entering the pipes leading to main condensers, auxiliary condensers, jet heaters, and the low-pressure receivers. These separators are located under lower deck in the engine rooms, one on each side above main air pumps.

The diameter of main steam pipes is 11 inches, and the diameter of steam pipe from each of the main boilers is 4 inches. The auxiliary steam pipes are eight inches in diameter, and the steam exhaust pipes are 10 inches in diameter.

All pipes subject to a steam pressure of 250 pounds per square inch are solid-drawn steel pipes, with fittings and flanges of crucible steel. All auxiliary steam pipes subject to a steam pressure of 150 pounds per square inch, and all feed pipes, are of solid-drawn copper with brazing metal flanges containing 80 parts of copper and 20 parts of zinc, with the exception of steam pipes to windlass and steam-steering gear,

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which are lap-welded wrought-iron pipes expanded and beaded into semi-steel flanges.

The deck fire and water-service pipes are lap-welded iron pipes, expanded and beaded into semi-steel flanges, which are faced after being secured to pipe, and then pipe and flanges are galvanized together. All steam-exhaust pipes are of solid-drawn copper, with flanges and fittings of semi-steel.

ELECTRIC PLANT.

The electric plant is located on orlop deck, at the forward end of engine rooms. It extends the full width of the vessel and is divided by the longitudinal bulkhead through which a vertical watertight door provides communication between port and starboard sides. There is installed on each side, three direct-connected Westinghouse generating sets, each of 75 kw., driven by a Westinghouse vertical compound engine. Each generator is a six-pole machine, furnishing direct current at 110 volts when making 300 revolutions per minute.

DATA FOR GENERATOR ENGINES.

Total number of engines	6
Diameter of H.P. piston, inches	11
L.P. piston, inches	19
Stroke, inches	11

The wiring is on the three-wire system, and several switchboards are conveniently placed throughout the vessel; lights take 110 volts, power and heating, 220 volts. The cables from each generator are led to the switchboards in the engine rooms and from these switchboards to a large switchboard in the central station on upper deck. Motor drive is used for the refrigerating machinery, cargo winches, ventilating fans, steering gear, laundry machinery, coal conveyor, coal elevator, provision hoist, machine shop and after warping capstans.

The electric-light installation consists of about 1,300 16-candlepower lights, distributed throughout the vessel, and a powerful 24-inch searchlight of the General Electric make, located on the forward lookout bridge. This light can be oper-

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ated from the navigating bridge by an electric or hand gear. The port, starboard and masthead lights are electric, and a tell-tale is fitted in the chart house, which indicates if lights are burning or not. For signalling in fog, an electrically-operated steam whistle is installed, which is automatically sounded at desired intervals by a clock control, and can also be controlled by hand.

There are two different telephone installations: one, the general, which is a regular telephone exchange connecting the various staterooms with the steward's office; the other is a special system which connects the executive offices of the vessel with each other. The switchboard of the general system is located in a separate room on the upper deck forward of the central electric station. Besides the instruments in the staterooms, telephones are installed in almost every part of the vessel, such as the Captain's private stateroom, the ladies' boudoir, library, smoking room, barber shop, kitchen, saloon pantry, laundry, chief engineer's office and chief electrician's office, so that no matter what the passengers want they can be connected with the department desired.

REFRIGERATING PLANT.

The refrigerating plant is located on the upper deck in the forward end of forward deckhouse and extends the full width of the house. The plant consists of two 30-ton Linde marine ammonia compressors with the necessary accessories.

The space for carrying refrigerated cargo has a capacity of about 650 tons, located in lower part of hold No. 3; it is divided into several compartments which are installed with mineral wool and sheathed with wood. In addition to the above, there is a space in hold No. 5, also divided into several compartments, where ship's provisions are stored and kept at varying temperatures, from 26 degrees Fahrenheit to 40 degrees Fahrenheit. The compartments for provisions are cooled by rows of brine pipes, and the cargo spaces are cooled by cold air which is driven through ducts leading from air coolers.

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There is also in connection with the refrigerating plant, an

ERRATA.

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Page 394, 10th line from bottom, change "650 tons" to read "1,650 tons."

each driven by a 7½-H.P. motor.

The drinking water is circulated through the cooler by a circulating pump driven by a 3-H.P. motor.

MACHINE SHOP.

The machine shop is located on 'tween deck on the outboard side of port engine-room casing, extends the full length of same, and is entered from the engine-room 'tween-deck grating. The following machines are installed:

One Prentice Brothers 30-inch upright drill;
One Prentice Brothers 14-inch sensitive drill;
One Flather 24-inch engine lathe, with taper attachment and compound rest;
One Hendy 24-inch shaper, with adjustable swivel table;
One 12-inch Chicago emery wheel grinder;
One 30-inch × 4-inch grindstone with truing device;
One power hack saw.

These tools are provided with the requisite attachments. Lockers and tool benches are also provided. The machines are driven by a 10-H.P. motor located on bulkhead at the after end of the machine shop.

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ANCHOR WINDLASS.

There is a vertical Hyde steam windlass and two warping capstans located on the upper deck forward, the engines being located on the forecastle deck below; the two vertical shafts are driven by worm gear direct from the worm on the crank shaft of the engine.

Each vertical shaft carries on its upper end, above upper deck, a wild-cat with locking gear. The arrangement is such that the wild-cats can be operated together or independently of each other. Each wild-cat is fitted with a positive locking device, worked by raised cams on the periphery of the locking rim and slotted block keys operated by means of levers. The entire operation of locking or unlocking the wild-cat is accomplished by one motion of the lever through an angle not exceeding 60 degrees. A friction-band brake is fitted to each wild-cat. Each friction band is provided with a compressor, so that either chain can be checked when running out.

The two warping heads, each about 26 inches diameter of barrel, are operated by the windlass engines through bevel and worm gearing. These warping capstans are capable of taking in rope at the rate of 10 fathoms per minute and to operate together or independently of each other. The engines are reversed by a change valve, so arranged as to be operated from both the upper and forecastle decks.

The windlass, as a whole, with 150 pounds of steam per square inch, will raise two $8\frac{1}{2}$-ton anchors with 60 fathoms of $3\frac{3}{16}$-inch chain at the rate of 5 fathoms per minute.

DATA FOR ENGINES.

Number of cylinders	2
Diameter of cylinders, inches	18
Stroke, inches	14
Diameter steam pipe, inches	5
exhaust pipe, overboard, inches	6
to condenser, inches	4

STEERING GEARS.

There is a Williamson Brothers combined steam and electric steering gear. Both are directly attached to the quadrant and

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are located on the upper deck in the steering-gear house aft. The engines and motor are controlled from steering columns on upper and lower navigating bridges by the same hand-steering wheel, a hydraulic telemotor, actuating the engine valves, while the electric steering motor is controlled through the Electro Dynamic Company's steering column on the lower navigating bridge. On the lower-deck platform, near the forward end of port-engine room, close to the longitudinal bulkhead, is located a motor-driven generator (Booster) with an exciter attached to the same shaft. The fields of the exciter are in circuit with the balancing wire connecting the steering resistance with the rudder resistance. By suitable arrangement and proportions of generator and exciter, to multiply the effect of a small current flowing in the balancing wire, sufficient power is obtained to drive the rudder motor.

A pair of warping capstans are located on the upper deck, one on each side of the steering-gear house aft; they are driven by a motor located on main deck.

DATA FOR STEERING GEARS.

Number of steam cylinders	2
Diameter of steam cylinders, inches	13
Stroke, inches	10
H.P. of electric steering-gear motor	75
motor for after warping capstans	100

DOCK TRIAL AND PRELIMINARY SEA TRIAL.

The main engines and auxiliary machinery were worked under steam several times before the S. S. *Dakota* left the builders' wharf, and on March 3, 1905, the official dock trial took place. On that day the main engines were worked for eight hours, up to 60 revolutions per minute, with only four boilers on each side.

At the request of the Great Northern Steamship Company the S. S. *Dakota* was taken to sea for a 24-hour run, in order to adjust compasses and give the machinery a trial under actual conditions at sea before leaving the hands of the builders.

On the return to New London, Conn., March 24, 1905, every-

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thing was opened up in engine and boiler rooms, inspected, and pronounced to be in a satisfactory condition, as the result of which the vessel was handed over to the owners.

At 1 P. M., on March 27, 1905, the S. S. *Dakota* left New London, Conn., for Newport News, Va., in charge of Mr. C. C. Lacey, the marine superintendent of the Great Northern Steamship Co., and Mr. Robert S. Paul, the chief engineer in charge of the engine and boiler rooms. The builders were represented by several gentlemen who had been in charge of the different departments during the construction and building of this vessel.

During the run to Newport News, Va., several observations were taken in connection with the machinery and boilers, particular care being taken in training the firemen to fire the water-tube boilers, and good results were obtained with a comparatively green crew of firemen, for that kind of boiler; this was due to the intelligent engineering staff, and in particular to the efforts of Mr. Andrew Bishop Cox, who represented the Stirling Company.

On this run only twelve boilers were in use, including the four boilers with mechanical stokers; the steam pressure in boilers was kept at about 200 pounds per square inch and at no time was it over 210 pounds with the induced-draft fans running from 150 to 175 revolutions per minute, the main engines making, on an average, 60 to 65 revolutions per minute.

The bottom of vessel, after being so long in the water, was naturally very foul, and the speed on that account not as it would have been with a clean bottom.

The water used on this trip for making up the boiler feed was carefully taken every hour, and was found to be on an average of 3.41 tons of 2,240 pounds per watch of four hours.

Steamship *Dakota* anchored at Hampton Roads, Va., at 12.15 A. M. on March 29th, 1905, and entered the dry dock at Newport News Shipbuilding and Drydock Co., 3 P. M. on the same day. While in the drydock the cast-steel propeller blades were taken off and manganese-bronze blades, set at a

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pitch of 22 feet, were put on. The bottom was given two coats of anti-corrosive and one coat of anti-fouling paint of the Rahtjen "Red Hand" brand.

OFFICIAL SPEED TRIAL.

S. S. *Dakota* left Newport News, Va., at 5·30 P. M. on April 1st, 1905, and steamed slowly to sea. Her draught on leaving Newport News was 23 feet aft and 15 feet 8 inches forward; but, on the way to sea, some of the tanks forward of midships were filled to get her on a more even keel, and the draught on trial was 21 feet 9 inches aft and 17 feet 1 inch forward, corresponding to a displacement of about 17,000 tons.

After midnight in the early morning of April 2d, 1905, the fire was started in all the main boilers, the steam pressure increased and the main engines gradually worked up to full power.

At about 5 A. M. she was turned around and started for a straight course between Winter Quarter Shoal lightship to Cape Charles lightship, off the coast of Virginia. The wind was about 18 knots, abaft beam, the tide about slack, if anything a little in her favor, and the sea was comparatively smooth. The time for passing the above-mentioned points was taken on the bridge by Captain Emil Francke, the commander of the vessel.

Passed Winter Quarter Shoal lightship	5·37 A. M.	
Cape Charles lightship	8·57 A. M.	
Elapsed time in hours and minutes	3 hours 20 minutes.	
Distance between two points, knots	57 (about).	
Average speed in knots per hour	17.1	
	Port. Eng.	*Star. Eng.*
Total revolutions during trial run	17,332	16,930
mean of both engines	17,131	
Slip of propeller in per cent. of its own speed	8.049	
Maximum I.H.P. developed	6,090	5,850
total both engines	11,940	

For further information, see accompanying tables giving results obtained on the speed-trial run.

The main engines during the trial ran smoothly with very little vibration, and none of the bearings heated. The internal water service was used in the main bearings, crosshead

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guides and thrust bearings, but no water was used on any other parts.

The main boilers made all the steam the engines could use, and the safety valves were blowing off steam practically the entire run, they being set at 250 pounds pressure. The mechanical-stoker boilers seemed to do more than their share, and after the speed trial the engineers spoke of the stoker boilers as good steamers, and the prejudice against the stokers commenced to disappear.

In using the pyrometer on top of the boiler tubes in stoker boilers the temperature varied but very little for the same speed of grate, but that can not be said about the boilers fired by hand, as the temperature above boiler tubes in those boilers varied considerable, due to the opening and closing of furnace doors.

I believe, basing my statement on what I saw of the mechanical stokers of steamship *Dakota*, that the mechanical-stoker principle for firing water-tube boilers on shipboard will sometime in the future take the place of hand firing, as it has done on larger boiler plants on shore.

The smoke emitted from the funnel during the trial was light gray nearly all the time, and at no time was it very heavy. The automatic feed-regulating valves on the main boilers worked almost without attention, and there was no trouble experienced with the feeding of the main boilers.

An attempt was made to find out the coal consumption during the trial, but failed on account of incomplete arrangement being made beforehand, and what preparation there had been made failed when the time came to try the correctness of the results. The coal consumption was, however, estimated to be 1.75 pounds per I.H.P. per hour. The kind of coal used was Pocahontas.

The water used for making up the boiler feed during the speed trial of three hours and twenty minutes, according to the float in the reserve feed tank, was about 41 tons, and, although the safety valves were blowing off steam a great deal, we knew that the water consumption could not have been as large as stated above, on careful inspection by the first assistant

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engineer it was found that the drains from grease extractors on the port main feed line had been left open by the man cleaning the filters the day before leaving Newport News, Va., which accounted for the great water consumption.

After the trial S. S. *Dakota* was headed for Norfolk, Va., to take on bunker coal at Lamberts Point coal piers, where she arrived at 3 P. M. on the same day.

Some trouble was experienced in coaling her on account of her great height, and it took several days to fill her bunkers with 5,226 tons of coal.

The *Dakota* left Norfolk, Va., at 10 A. M. on April 12th and arrived at the pier in New York at noon on April 13th, 1905.

In New York, while taking in a cargo of over 9,000 tons dead weight, consisting principally of steel rails, water and provisions for her long voyage, the main engine cylinders, valve chests, low-pressure valves, crank and crosshead-pin bearings, crank-shaft bearings and crosshead guides were examined and found to be in excellent condition.

In taking off the covers on main condenser heads the tube-packing glands and nuts on tube-sheet braces were found to be a little loose. They were all set up and tested for tightness. The main air pumps and main feed pumps were also opened up and found fairly free from grease and in good condition.

All the manhole covers on steam drums of main boilers were taken off, drums examined, and there was no sign of any grease in same, with the exception of in boilers Nos. 1 and 2, which had been used as donkey boilers at Norfolk, Va.; in these was found a very light coat of grease at the water line. Several tubes were taken out in the different boilers, examined and found free from grease and in good condition. No difficulty was encountered in removing the tubes.

Steamship *Dakota* left New York at 1·30 P. M. on April 28th, 1905, drawing 32 feet aft and 31 feet 6 inches forward. The first stop is supposed to be Coronel, Chili, and I know of no reason why she should not make a creditable showing on the run to Seattle, Washington.

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RESULTS OF SPEED TRIAL OF S. S. *DAKOTA*.

Date, April 2, 1905. **Table No. 1.** Eastern Shipbuilding Co., New London, Conn.

Time.		Opening of throttle.		Position of main valve links.		Readings of counter.		Revolutions per minute.		Steam pressure per square inch (above atmosphere). Main steam at throttle.		H.P. steam chest.		M.P. receiver.		L.P. receiver.		Auxiliary steam.		Vacuum. Main condenser.	
P.	*S.*	*P.*	*S.*	*P.*	*S.*	*P.*	*S.*	*P.*	*S.*	*P.*	*S.*	*P.*	*S*	*P.*	*S.*	*P*	*S.*	*P.*	*S.*	*P.*	*S.*
A.M.	A.M.	*Inches.*	*Inches.*															*Inch.*	*Inch.*	*Inch*	*Inch.*
5.30	5.30	.66	.68	Normal: 10-inch travel of valves.	Normal: 10-inch travel of valves.	60,920	65,660	83	82	235	235	215	215	62	62	10	10	150	150	24	24
5.37	5 37	.....	.....			61,518	66,240	...	Winter quarter shoal light ship abeam.									...	...	...	...
6.00	6.00	.68	.70			.....	.....	83	82	235	235	215	215	62	62	10	10	150	150	24	24
6.30	6 30	.7	.75			.....	.....	84	83	236	236	218	218	63	63	10¼	10½	150	150	23½	23¾
7.00	7 00	1.	1.			.....	.....	85	84	236	236	218	218	70	71	10¼	10½	150	150	23½	23¾
* 7.30	7.30	2.	2.			.....	.....	88	87	240	240	225	220	70	80	13	13	150	150	23	23½
8.00	8.00	2.	2.			.....	.....	90	89	240	240	225	220	70	82	14	14	150	150	22½	23
8.30	8 30	1¾	1¾			.....	.....	87	86	238	238	220	220	70	82	14	14	150	150	22½	23
8.37	8 37	1½	1½			78,850	83,170	...	...	...	Cape Charles light ship abeam.							...	...	...	...
9.00	9.00	1½	1½			.....	.....	84	83	235	235	215	215	70	80	13	13	150	150	23	23½

* M.P. and L.P. pass-over valves open during last five periods. Auxiliary condensers not used. No steam pressure in cylinder jackets during speed trial.

Table No. 2.

Time.		H.P. Cylinder. Number of card		Scale of spring.	M.P. cylinder. Number of card.		Scale of spring	L.P. Cylinder Number of card.		Scale of spring.	Gauge pressures, in pounds per square inch. Jet heater.		Feed-water filter.	
P.	*S.*	*P.*	*S.*		*P.*	*S.*		*P.*	*S.*		*P.*	*S.*	*P.*	*S.*
A.M.	A.M.													
5.30	5 30	1	1	100	1	1	50	1	1	16	4	4½	265	265
6 00	6.00	2	2	100	2	2	50	2	2	16	4	4½	266	266
6 30	6.30	3	3	100	3	3	50	3	3	16	4½	5	268	270
*7 00	7.00	4	4	100	4	4	50	4	4	16	4½	5	262	263
7.30	7.30	5	5	100	5	5	50	5	5	16	4½	5¼	264	265
8.00	8 00	6	6	100	6	6	40	6	6	16	5	5¼	268	271
8 30	8.30	7	7	100	7	7	50	7	7	16	5½	5½	266	268
9.00	9 00	8	8	100	8	8	50	8	8	16	5½	5½	263	264

* M.P. and L.P. pass-over valves open during last five periods.

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RESULTS OF SPEED TRIAL OF S. S. *DAKOTA*.

Date, April 2, 1905. Eastern Shipbuilding Co., New London, Conn.

Table No. 3.—PORT.

Time.	Steam pressure given in pounds per square inch above atmosphere. vacuum given in inches of water.																Temperature, degrees Fahrenheit.													
	Steam pressure in main boilers, pounds.								Vacuum in ash pits, inches.										In ash pit.								Of gases.			
																											Above boiler tubes.	Above heater tubes.	Above boiler tubes.	Above heater tubes.
	No. 1.	No. 2.	No. 7.	No. 8.	No. 9.	No. 10.	No. 15.	No. 16.	No. 1.	No. 2.	No. 7.	No. 8	No. 9	No. 10.	No. 15.	No. 16.	After fire room.	Forward fire room.	No. 1.	No 2.	No. 7.	No. 8.	No. 9.	No. 10.	No. 15.	No. 16.	Blr. No. 10.		Blr. No. 16.	
A.M.																														
5 30	250	250	250	250	250	250	250	250	.25	3	.2	.3	.25	.2	.3	.4	120	106	300	280	260	265	280	275	300	320	730	550	720	542
6.00	250	250	250	250	250	250	250	250	.3	.2	.3	.2	.2	.25	.2	.4	122	106	310	280	275	260	300	280	320	330	735	555	718	541
6 30	250	250	250	250	250	250	250	250	.2	.2	.3	.25	.1	.3	.2	.5	123	106	324	320	395	270	296	285	312	310	740	560	725	546
7 00	245	245	245	245	245	245	245	245	.25	.25	.35	1	.2	.15	25	.5	122	110	322	300	300	275	296	288	312	310	750	570	730	552
7 30	248	248	248	248	248	248	248	248	.3	.3	3	.2	.25	.2	.3	.5	132	115	324	320	310	285	306	290	315	315	745	560	732	552
8.00	250	250	250	250	250	250	250	250	.2	.2	.25	.25	.2	.75	.25	5	130	118	330	325	308	290	310	285	320	310	740	560	730	551
8.30	248	248	248	248	248	248	248	248	.25	25	.3	.22	.22	.3	3	.5	130	120	330	320	310	292	312	290	322	320	725	550	728	550
9.00	245	245	245	245	245	245	245	245	.3	.3	.35	.2	.25	.2	.25	.5	130	120	330	320	310	295	312	295	319	310	730	552	725	552

Table No. 4.—STARBOARD.

	No. 3.	No. 4.	No 5.	No. 6.	No. 11	No. 12.	No. 13.	No. 14.	No. 3.	No. 4	No. 5.	No. 6.	No 11.	No. 12.	No. 13.	No 14.			No. 3.	No. 4.	No. 5.	No. 6.	No. 11.	No. 12.	No. 13.	No. 14.	Boiler No. 11.		Boiler No. 4.	
A.M.																														
5.30	250	250	250	250	250	250	250	250	.25	.5	.25	.25	.3	.2	.25	.25	120	96	280	290	285	290	252	210	215	225	735	555	710	535
6.00	250	250	250	250	250	250	250	250	.25	.5	.4	.25	.5	.25	.25	.25	125	96	280	290	290	270	250	215	218	230	740	560	710	535
6.30	250	250	250	250	250	250	250	250	3	.5	.3	.25	.5	.25	.25	.25	130	98	280	300	295	272	250	215	218	232	745	566	720	540
7.00	245	245	245	245	245	245	245	245	.4	.5	.5	.25	.5	25	.25	.25	130	100	300	310	300	275	260	215	220	230	745	566	725	546
7 30	248	248	248	248	248	248	248	248	.4	.5	.5	.25	.5	.25	.25	.25	135	102	304	320	310	280	275	220	225	224	750	570	728	550
8.00	250	250	250	250	250	250	250	250	.5	.5	.5	.25	.5	.25	.25	.25	135	104	306	322	310	282	275	220	220	225	748	568	730	551
8.30	248	248	248	248	248	248	248	248	.5	.5	.4	.25	.5	.25	.25	.25	136	101	304	322	308	280	278	218	218	230	745	565	728	549
9.00	245	245	245	245	245	245	245	245	.4	.5	.3	.25	.5	.25	.25	.25	136	100	300	320	306	280	270	218	220	225	740	560	725	545

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RESULTS OF SPEED TRIAL OF S. S. *DAKOTA*.

Date, April 2, 1905. Eastern Shipbuilding Co., New London, Conn.

Table No. 5.

Time.		Revolutions, or double strokes, per minute.										Temperature, degrees Fahrenheit.									
		Main circulating pump.		Air pump.		Main feed pump.		Main feed lift pump.		Auxiliary circulating and air pump.		Engine-room working plate.		Main circulating pump. Suction.		Main circulating pump. Discharge.		Hot wells.		Feed water leaving heater.	
P.	*S.*	*P.*	*S.*	*P.*	*S.*	*P.*	*S.*	*P.*	*S.*	*P.*	*S.*	*P.*	*S.*	*P.*	*S.*	*P.*	*S.*	*P.*	*S.*	*P.*	*S.*
A.M.	A.M.																				
5 30	5.30	190	210	46	28	24	20	21	18	Not used during trial.	Not used during trial.	70	74	45	45	87	88	62	76	170	178
6 00	6 00	190	190	46	50	24	20	21	18			70	74	45	45	87	88	62	76	170	178
6.30	6.30	187	190	46	50	24	20	21	18			70	74	45	45	87	88	62	76	180	182
7.00	7.00	187	190	46	24	26	20	21	20			70	74	45	45	87	88	62	76	182	183
7.30	7.30	185	185	44	24	24	19	21	19			71	75	46	45	87	90	62	78	182	188
8 00	8.00	184	185	44	26	26	28	20	20			72	76	45	45	93	97	66	85	192	195
8.30	8.30	184	185	44	26	26	28	20	20			72	76	45	45	93	98	66	86	198	200
9.00	9.00	190	185	44	26	24	21	21	21			72	76	45	45	93	96	66	84	198	200

Table No. 6.

Time.		Revolutions of induced-draft fans.				Steam pressures at fan engines.		Temperatures at fan inlets.				Temperatures in fan-engine rooms.				Vacuum in inches of water at fan inlets.			
		Port.		Starboard.				Port.		Starboard.		Port.		Starboard.		Port.		Starboard.	
P.	*S.*	Aft. No. 1.	For'd. No. 2.	Aft. No. 3	For'd. No. 4.	*P.*	*S.*	Aft No. 1.	For'd. No. 2.	Aft. No. 3	For'd. No 4.	Aft. No. 1.	For'd. No. 2	Aft. No. 3.	For'd. No. 4.	Aft. No. 1.	For'd. No. 2.	Aft. No. 3.	For'd. No. 4.
A.M.	A.M.					*Lbs.*	*Lbs.*	*Deg.*	*Deg.*	*Deg.*	*Deg.*	*Deg.*	*Deg.*	*Deg.*	*Deg.*	*Inches.*	*Inches.*	*Inches.*	*Inches.*
5.30	5.30	180	180	190	180	150	150	545	540	590	510	78	75	80	75	1.2	1.	1.3	0.6
6 00	6.00	180	180	190	180	150	150	550	500	600	500	78	76	81	76	1 1	0.9	1 2	0.6
6 30	6.30	180	180	200	180	150	150	540	500	600	490	79	78	82	77	1.	1.2	1.2	1.
7.00	7.00	200	200	200	180	150	150	560	530	580	475	80	79	82	78	2.	1.	1.4	1.5
7.30	7.30	200	180	200	190	150	150	565	535	570	505	80	80	83	79	1.7	1.5	1.5	1.8
8.00	8.00	200	190	200	200	150	150	600	550	615	520	81	82	84	80	2.	1.6	2.	1.8
8 30	8.30	190	180	200	200	150	150	555	560	595	530	81	82	85	81	1.7	1.4	1.4	1.7
9 00	9.00	200	200	190	200	150	150	560	545	550	445	81	82	85	81	2.	2.2	1.6	2.

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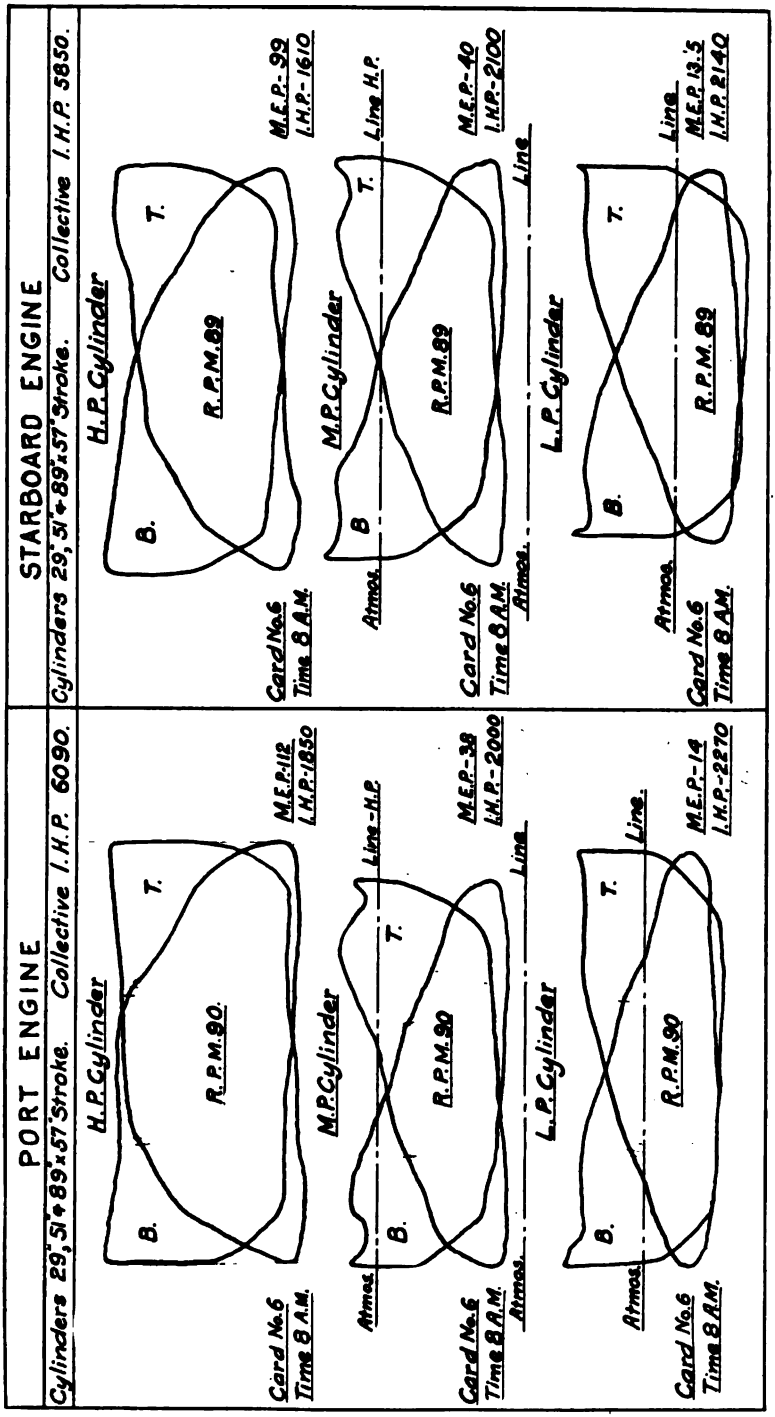

MAIN ENGINE INDICATOR CARD S. S. "DAKOTA."

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TEST OF EVAPORATORS, U. S. S. *DIXIE*.

FROM A REPORT MADE BY LIEUTENANT NEWTON MANSFIELD, U. S. N., MEMBER.

DESCRIPTION OF PLANT.

There are three evaporators, rated capacity, 4,000 gallons per diem, single nest, straight horizontal-tube Navy type. All are fitted to take steam direct from the boilers, each having a trap to return this condensed steam to the feed tank in the engine room.

There are three straight-tube vertical distillers—two of Williamson Brothers' patent, of 3,750 gallons per diem each, and one vertical straight-tube Navy type, of 2,500 gallons.

One horizontal 9 by 8 by 12 Davidson pump furnishes circulating water.

One horizontal 5½ by 5 by 5 Davidson pump feeds the evaporators.

One vertical, single-cylinder, Blake pump, 4½ by 2¾ by 6, delivers the product of the distillers to two test tanks of 500 gallons each.

There are two filter tanks which are not used, except to catch the distilled water as it comes from the distillers.

The three evaporators are known as "A," "B" and "C." On December 24 to 27, 1903, an inch and one-half pipe was fitted to carry the product of "A" to the coils of both "B" and "C." A larger pipe, which was considered necessary, was not fitted owing to the limited time which the ship remained at the League Island Navy Yard. A separate connection was made from the traps of "B" and "C" to the distillers so that the product of "A" entering the coils of "B" and "C" could be trapped to the distillers, there cooled, pumped to the testing tanks and measured.

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ARRANGEMENT OF THE TESTS.

The tests are divided into three stages, viz:

First Stage.—Running the plant double effect, steam being taken from the boilers only in the coil of "A," the product of all three evaporators being carefully measured in the test tanks.

"A" was blown down once every four hours, "B" and "C" once every twelve hours.

The pressure maintained in "A" was, coil, 55 pounds, shell, 35 pounds. That in "B" and "C," coils, 18 pounds, shells, 8 pounds. The speed of the circulating pump was 38 double strokes per minute.

Second Stage.—"A" was operated single effect, steam being taken into the coils of this one evaporator from the boiler, and returned to the boiler by the way of the trap and engine-room feed tank. The product was carefully measured. "A" was blown down once every four hours. The pressure maintained in "A" was, coils, 38, shell, 19. One distiller of 3,750 gallons capacity was found to be insufficient to condense the steam, so a second one was used, thus showing that the distillers are too small for the capacity of the plant when working single effect. The speed of the circulating pump was necessarily increased to 60 double strokes a minute.

Third Stage.—All evaporators and evaporator pumps being shut down, the coal consumed for all other auxiliaries in the ship for three days was carefully tallied.

CONDUCTING THE TESTS.

The following conditions were maintained as nearly as possible, viz:

Each stage was to continue for three days.

The auxiliaries in the whole ship were carefully noted, and an endeavor was made to have the time each one was in operation during each stage of the test equal that of each of the other two stages.

All the coal, excepting that for the galley and steam launch,

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was carefully tallied for each stage. A number of buckets were weighed and an average obtained.

The first and second stages were carried on to completion, but the third stage was interrupted at the end of the fortieth hour by orders to get under way.

RESULTS.

Table I gives the number of hours that each auxiliary was in operation during each stage; also the total coal consumed by the boiler during each stage. Column IV is made up for 72 hours from column III, which is for 40 hours only. Column IV is then taken as the consumption of coal for all the auxiliaries during each of the three stages, as the auxiliaries in their time of operation varied but little.

Table I.

—	Stage I.	Stage II.	Stage III.	
	72 hours. I.	72 hours. II.	40 hours. III.	72 hours. IV.
Evaporators "A," "B" and "C," double-effect, hours	72	0	0	0
Evaporator "A," single-effect, hours	0	72	0	0
Evaporator pumps, hours	72	72	0	0
Auxiliary condenser, hours	72	72	40	72
feed pump, hours	72	72	40	72
Flushing pump, hours	72	72	40	72
Ice machine (one), hours	72	72	40	72
One dynamo,* hours	73	73	43	77
Fresh-water pump, hours	20	21	9	16
Fire pump, hours	16	12	4	7
Machine-shop engine, hours	4	8	5	9
Deck winches, hours	2	0	0	0
Steering engine, hours	1	0	0	0
Total coal consumed, pounds	47,752	45,110.4	12,700	22,860
Water produced, gallons	21,000	10,000	0	0

* Two dynamos were in operation 1 hour.

Subtracting, then, column IV from column II, we have—

(1) Coal consumed by "A," single effect, in 72 hours, pounds.... 22,250.4
(2) Water produced by "A," single effect, in 72 hours, gallons.. 10,000
(3) Pounds of coal to produce one gallon of water, single effect.. 2.225

and subtracting column IV from column I, Table I, we have—

(4) Coal consumed by "A," "B" and "C," double effect, in 72 hours, pounds.... 24,892

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(5) Water produced by "A," "B" and "C," double effect, in 72 hours, gallons	21,000
(6) Pounds of coal to produce one gallon of water, double effect	1.185

Comparing (3) and (6)—

Saving in coal to produce one gallon of water by running double effect, pounds	1.045
The expenditure of water for ship's use is in one month, gallons	98,500
make-up feed for one month, gals...	9,000
in one month, gallons	107,500
Saving in coal in one month by using double effect, pounds	112,337.5
tons	$50\frac{837}{1240}$
year by using double effect, tons	$601\frac{1804}{1240}$
At $4.50 per ton, saving in one year in cost of operation	$2,708.12

CONCLUSION.

To further the fighting efficiency as well as the operating efficiency, all of our naval vessels should have their evaporating plants operated on the double-, or better still, the triple-effect systems.

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THE TURBINE STEAMSHIP *VICTORIAN.*

REPORT MADE BY COMMANDER A. B. CANAGA, U. S. N., MEMBER.

In obedience to the Department's orders of April 1, 1905, I proceeded to Halifax, Nova Scotia, and made the passage to Liverpool, England, and return on the turbine steamer *Victorian* of the Allan Line, plying between Liverpool and Canadian ports, in order to observe the performance of the turbine machinery.

It is a matter of regret that my orders required me to make the return trip by the first turbine steamer sailing, which was the *Victorian*, as an inspection of the operation of the turbines of the sister steamer, the *Virginian*, would have been particularly interesting and valuable, since these turbines were constructed by the Parsons Marine Steam Turbine Company, while those for the *Victorian* were constructed from Parsons' plans by Workman, Clark & Co., of Belfast, the builders of this vessel.

The managers, agents and ship's officers were courteous and obliging, and I was allowed to examine the arrangement of all machinery and to inspect freely its operations, both while maneuvering and when in free route.

DIMENSIONS OF S. S. *VICTORIAN.*

Length, feet,	540
Beam, feet,	60
Draught, feet,	27
Tonnage, gross, tons,	11,200

The draught at Liverpool was 27 feet 6 inches, aft; was unable to obtain draught forward. At Quebec, after making the trip from Liverpool, the draught was 23 feet 6 inches forward and 26 feet 6 inches aft. The draught as given was taken

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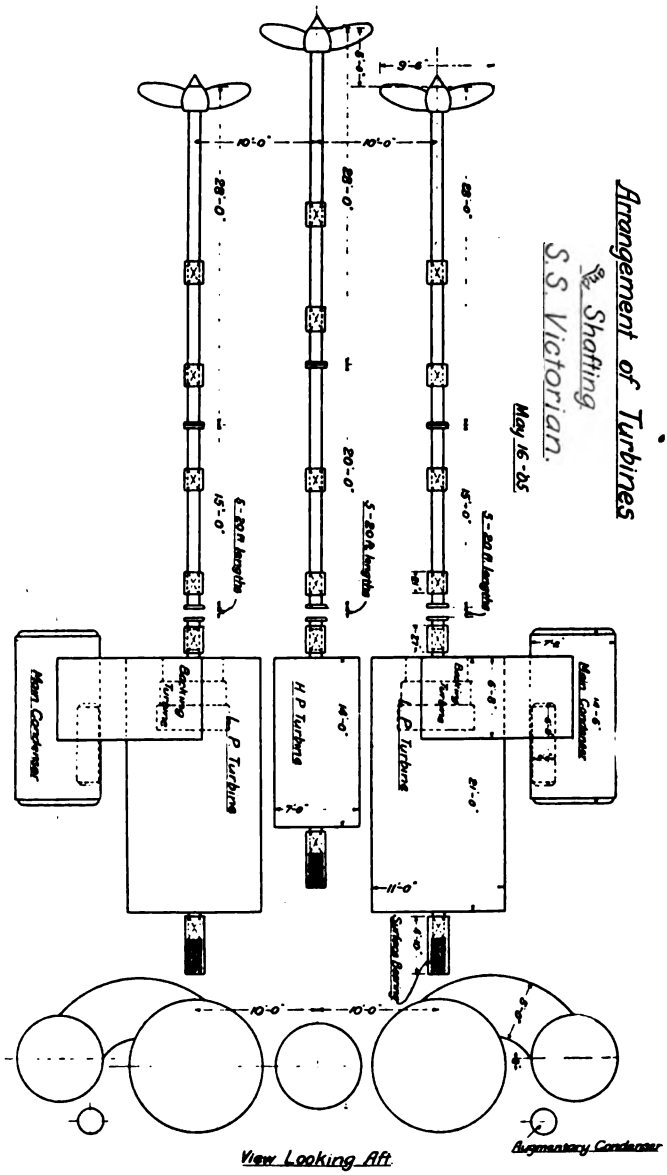

Fig. 1.

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in brackish water and after the vessel had completed the voyage from Liverpool, and it was, of course, greater than would have been shown with same displacement in open sea.

BOILERS.

There are eight cylindrical boilers of the "Scotch" type constructed for working pressure of 180 pounds per square inch.

Three double-end boilers:	
Diameter, feet	16
Length, feet	20
Three single-end boilers:	
Diameter, feet	17
Length, feet	12
Two single-end boilers:	
Diameter, feet and inches	14–2⅝
Length, feet and inches	11–6
Total grate surface of all boilers, square feet	797.5
heating surface of all boilers, square feet	31,478
H.S. ÷ G.S	39.4

There are four furnaces in each single- and eight in each double-end boiler.

Length of grates, feet	5
Width, feet	3
Diameter of boiler tubes, inches	2¼

Boiler tubes are fitted with retarders.

There is one elliptical smoke pipe for all boilers; outside casing, 16 feet by 19 feet 6 inches.

The fire rooms, two in number, are arranged athwartships. The furnaces of three single-end and the forward furnaces of the three double-end boilers are worked from the forward fire rooms, while the after furnaces of the double-end and the furnaces of the after two single-end boilers are worked from the after fire room.

The original design contemplated three single and three double-end boilers. This was changed during construction and two additional single-end boilers were installed in space originally allotted to pumps and other auxiliaries.

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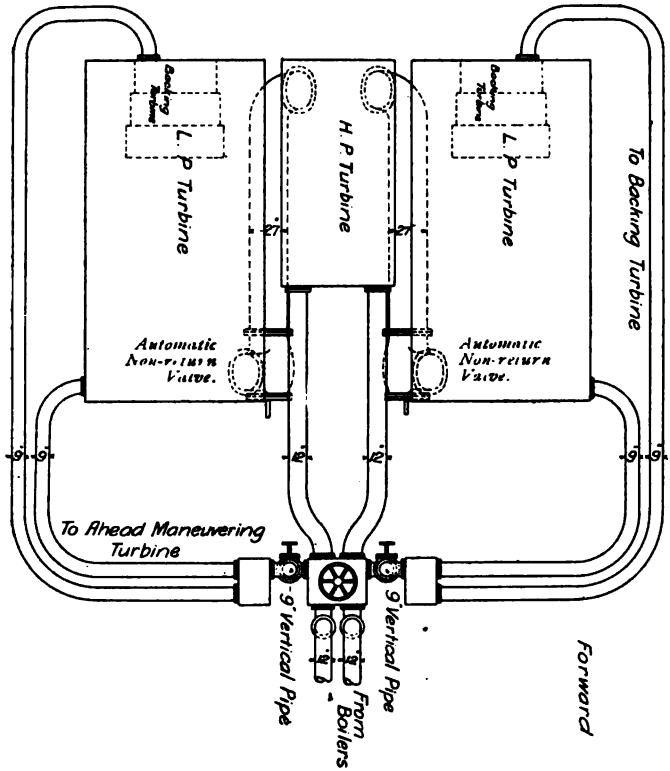

Fig. 2.

BLOWERS.

The boilers are fitted with hot-air draft of the Howden system. The air heaters are located in the uptakes. The blowers

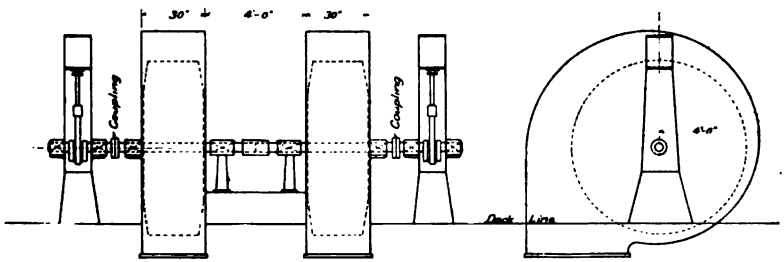

Fig. 3.

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are on the orlop deck in roomy spaces, supplying the air at a pressure of $2\frac{3}{4}$ inches to 3 inches in the blower casing and $\frac{3}{4}$ inch to 1 inch in ash pits. The temperature of the air is raised to 190 degrees Fahrenheit.

The blowers are arranged in pairs, each pair fitted with two engines, each having sufficient power to operate the two blowers. One engine of each pair is disconnected and kept in reserve. The arrangement of coupling the blowers in pairs is shown in Fig. 3.

COAL BUNKERS.

The bunkers have a capacity of 2,400 tons. The main bunker is forward of the boiler compartment. There are wing bunkers abreast the two after boilers, and these connect with hanging bunkers at the sides of forward half of engine room.

MAIN FEED PUMPS.

There are two main feed pumps of the Weir pattern $\frac{14 \times 19}{30}$ inches, making 15 to 18 double strokes per minute. The long stroke of these pumps and the slow movement make them specially reliable in their operation. The feed water is taken from the hot wells and forced through the feed-water heater, where its temperature is raised to 180 degrees Fahrenheit by the exhaust steam from the pumps and other auxiliaries before being fed to the boilers. The condensed steam from the feed-water heater, which is located overhead, is drained back to the hot wells. The remaining uncondensed exhaust of the auxiliaries is passed on to the auxiliary condenser, where it is condensed and drained by gravity to the hot wells. The auxiliary condenser is not fitted with an air pump, and, in consequence, there is a pressure at all times of five to six pounds per square inch in the exhaust pipes of the auxiliaries and in the feed heater.

MAIN CONDENSERS.

The total cooling surface is 17,000 square feet, divided between two main condensers of 8,500 square feet each.

Diameter of tubes, $\frac{3}{4}$ inch; thickness, No. 16 L.S.G.

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MAIN AIR PUMPS.

There is a duplex air pump of the Weir pattern for each condenser. Size, $\frac{11 \text{ inches} \times 31 \text{ inches.}}{21 \text{ inches.}}$ These air pumps, assisted by the Parsons intensifiers, were able to maintain a vacuum of 27 inches to 27½ inches when making 38 double strokes per minute.

INTENSIFIER.

The intensifier, an invention of Mr. Parsons, is an apparatus similar in action to the steam ejector. Its function is to remove the air and vapor from the condenser and deliver them direct to the air-pump suction, thereby assisting in maintaining the high vacuum so necessary to the economic performance of the turbine.

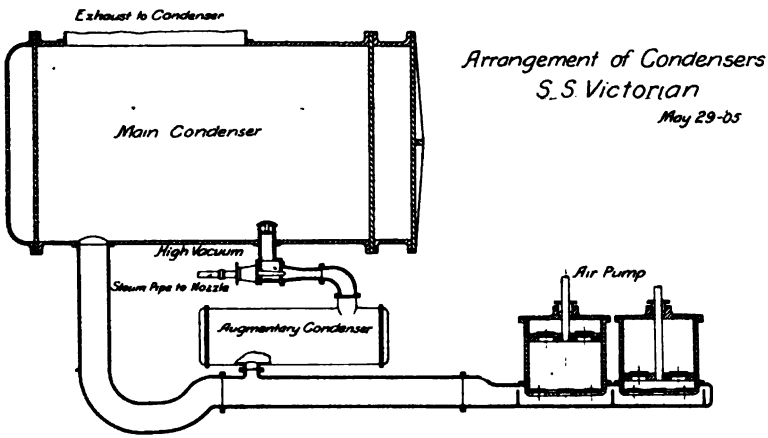

Fig. 4.

The steam jet carries the air and vapor through the intensifier to the augmentary condenser, where it is delivered to the air-pump suction near the air pump.

The location and arrangement of the intensifier and augmentary condenser with reference to the main condenser and air pump is shown on Fig. 4.

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CIRCULATING PUMPS.

The circulating water for each condenser is supplied by a centrifugal circulating pump having 20-inch suction and discharge. The condensing water enters the condensers at the top, returns through the lower tubes and is discharged from the bottom of the water end.

Each centrifugal circulating pump receives its water through two 15-inch sea valves, the 15-inch pipes from these sea valves uniting into a main injection pipe 20 inches in diameter. There is also a 10-inch bilge-injection pipe fitted to each of these pumps. Each centrifugal pump is operated by two single engines, each of sufficient power to operate the pump at 170 revolutions per minute. The object in having the two independent engines fitted to each pump is to admit one engine being kept in reserve. The arrangement of pump and engines is shown on Fig. 5.

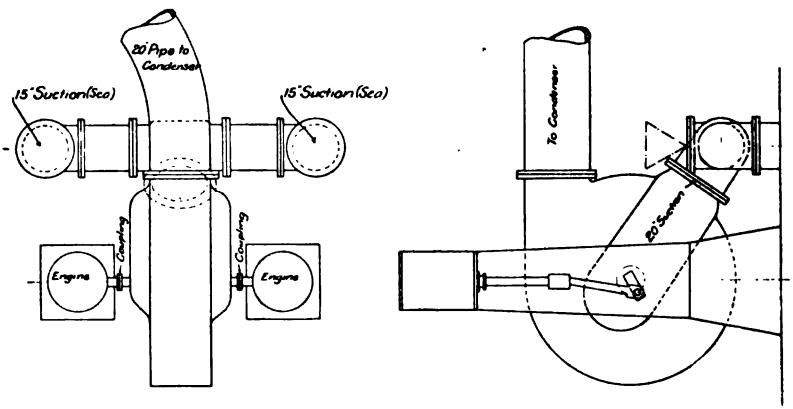

Fig. 5.

OIL PUMPS.

There are two oil pumps of about $\frac{4 \times 5}{6}$ inches supplying forced lubrication to the main bearings of the turbines One of these pumps is kept in reserve ready to be started at a moment's notice. As it is impossible to run the turbines without forced lubrication for their bearings, these pumps are

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made of ample size and located conveniently at the working platform, forward end of the H.P. turbine. A pressure of 10 pounds is maintained continuously in the oiling circuit.

AUXILIARY PUMPS.

There are a number of auxiliary pumps for feeding the boilers, for supplying condensing water to the auxiliary condenser, for operating the ash ejectors, for pumping feed water from the reserve tanks, for pumping hot and cold water to the galleys and passengers' lavatories, for pumping out double bottoms, etc.

REFRIGERATING MACHINERY.

There is a carbon di-oxide refrigerating machine of about twelve tons capacity on a flat over the engine room used for refrigerating purposes alone.

DYNAMO MACHINERY.

There are two direct-connected dynamo sets of 60 kw. each generating current for lighting ship and for operating the dough-kneading machine, dish-washing machine, revolving spits and the Sirocco ventilating blowers.

DECK AUXILIARIES.

The anchor, winch and steering engines are all operated by steam.

PROPELLERS.

There are three propellers, all the same size, and having four blades each.

Diameter, feet and inches	9– 6
Pitch, feet and inches	8– 6

I was unable to learn the area of the propeller blades.

TURBINES.

The steamship *Victorian* is fitted with turbines built by Workman, Clark & Co., shipbuilders and engineers, of Belfast, under license from the Hon. C. A. Parsons and the Parsons Marine Steam Turbine Company.

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The turbines for the sister ship *Virginian* were built from the same plans, but by the Parsons Marine Steam Turbine Company. There is a prevailing belief that the turbines of the *Virginian* are better, and more carefully built, and are already showing their superiority. A memorandum of the daily runs on the first trip from Liverpool to Halifax showed that the *Virginian* had made somewhat better speed than her sister, the *Victorian.*

There are three turbines, one high-pressure operating the central and two low-pressure operating the wing propellers. The central and starboard screws are right-hand; the port screw is left-hand.

The line shafting for each turbine is made up of six sections, each 20 feet long, except the after sections of the wing turbines, which are 15 feet in length. The propeller shafts are of the same length and interchangeable. The wing propellers are located 5 feet forward of the central propeller, as shown on Fig. 1. The shafts are parallel and spaced 10 feet center to center. The line and propeller shafting are forged solid and are rough-turned, 11½ inches in diameter, except at journals, where they are smoothed and polished.

The line shafting for each turbine is supported by fifteen bearings, 21 inches in length. Diameter of the journal in bearings same as diameter of the shafting. The line shaft bearings are fitted with oil rings, and are kept cool by water circulation through the bearings.

All main bearings of the turbines, both forward and aft, are 15 inches in diameter by 20 inches in length. These bearings are fitted with forced lubrication. The lubrication of these bearings is so important that it is necessary to have the system duplicated, maintaining at all times one oil pump in reserve.

The main and line shaft are all fitted with closed-circuit water circulation, one circuit taking water from the top of the forward water end of port condenser leads to all the line shaft bearings, the other leading from the top of the forward water end of the starboard condenser to the main bearings of the turbines. The two circuits unite at a small auxiliary pump

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which forces the water from the bearings through the oil cooler and on to the deck for sanitary purposes.

The oil in the forced-lubrication system is passed through the cooling coils and thence to a settling tank, from where it is pumped again to the bearings. There is a small waste of oil in the forced-lubrication system, and this is made up from time to time by fresh oil. The temperature of the oil in the forced-lubrication system remained at a temperature of 70 to 72 degrees Fahrenheit.

MAIN STEAM PIPING, FIG. 2.

The steam from the boilers is led into the engine room through two 12-inch steam pipes uniting in the throttle valve

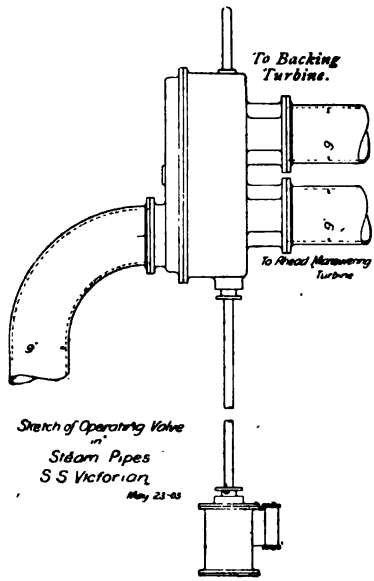

Fig. 6.

at the working platform. From the throttle valve the steam is led through two 12-inch pipes to the H.P. turbine. When in free route the steam is passed through the H.P. turbine where it separates, half going to the starboard and half to the port turbine through 27-inch receiver pipes, and from thence

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through exhaust pipes 6 feet 8 inches × 5 feet to the main condensers. For maneuvering, the main throttle is closed and steam admitted to the maneuvering valves, one for each L.P. turbine. The operating or maneuvering valves on the *Victorian* are worked by a steam cylinder shown on Fig. 6. They are simply slide valves, which, when placed at the upper end of stroke, admit live steam to the forward end of the low-pressure turbine, when at the bottom end of stroke admit live steam to the backing turbine, and when in mid position shut the steam from both the ahead and backing turbines.

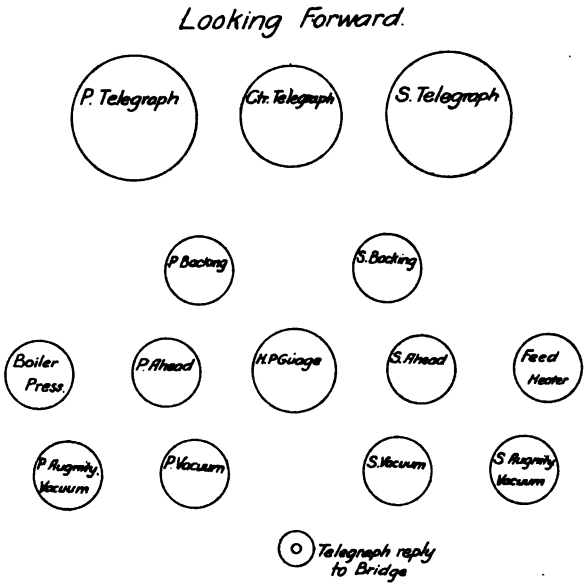

Fig. 7.—GAUGE BOARD.

In order to prevent the steam blowing back into the H.P. turbine when maneuvering, non-return valves are fitted in the receiver pipes between the H.P. and L.P. turbines, as shown on Fig. 4. These valves are automatic, opening or closing as the live steam is admitted to the H.P. or L.P. turbines.

At forward end of each turbine shaft is fitted a safety governor, which, in case of accident to propeller or shafting, operates to close the main throttle valve. It was noticed that

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while handling the turbines these governors served another purpose, indicating whether the turbines were at rest or in motion. From the working platform these governors were the only visible moving parts of the turbines.

The counters register one revolution for ten revolutions of the turbines. The gear for operating the counters is located on the shafting just abaft the after main bearings.

OPERATION.

As a rule the turbines are easily and quickly handled, the change being made from the conditions of free route to conditions for maneuvering by simply closing the main throttle and opening the maneuvering stop valves. The action of the maneuvering valves is instantaneous, and the turbines respond immediately to the signals. It is noted that the minor mishaps and annoyances met with in a reciprocating engine were to be found with the turbines.

Heating of the main bearings of turbines is, however, a more serious matter than with the reciprocating engines, as the melting of the white metal or unusual wear in these bearings would cause a serious interference between the rotors and casing. Ample surface of main bearings and careful lubrication reduce the chances of such heating.

After being under way three or four hours from Liverpool, the after main bearing of the port L.P. turbine heated, making it necessary to stop this turbine and proceed at a reduced speed with the other two. Owing to the rolling of the ship, some grit that had settled in oil reservoir was carried to the bearing surface, causing the heating. The trouble was soon discovered, so that little damage was done beyond a delay of about twenty hours at Moville while overhauling and cleaning the heated bearing. It was noted that during the remainder of the trip this bearing gave no further trouble.

There was some annoyance due to priming of boilers and water being carried over to the turbines from time to time. This, however, is not the same menace to the turbines as to reciprocating engines, when water carried over into the cyl-

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inder is liable to cause damage to the cylinder heads and pistons.

It is anticipated that this difficulty from priming, in a great measure, will disappear as the boiler plates and tubes lose the mill scale.

There was a pleasing absence of vibration, not only in the engine room, but throughout the ship, owing to the rotary motion and the perfect balance of the turbines. The propellers are also perfectly balanced, and, having a high speed of rotation, allow a corresponding reduction in their diameters. The small diameter admits of a much deeper immersion and consequent absence of racing. It was impossible to detect the slightest variation in the rotation of the turbines when the vessel was pitching in a heavy head sea. It is estimated that the tips of the blades at the highest points of their paths are still immersed 14 feet, which would require a degree of pitching rarely or never experienced before the propellers would be released sufficiently to allow of racing. This security from racing is a comfort to the passengers as well as to the engineers, as many of the troubles and mishaps to the propelling machinery come when the engines are racing and the vessel is straining and laboring in heavy seas.

Against this immunity from racing, due to the small diameter of screw with great immersion, must be set its lack of holding power. It was observed that the influence of head winds and seas reduced the vessel's speed considerably more than would have been the case with the large propellers used with reciprocating engines.

ENGINE-ROOM VENTILATION.

The radiation and leakage from the turbines and their connections makes the working platform and other working spaces in the engine room hot and uncomfortable, even though provision was made for the supply of fresh air by two 30-inch ventilators and for the escape of the hot air by a trunk 10 feet by 12 feet leading to the bridge deck. The after forced-draft blowers draw a part of their supply from the engine-room hatch

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at a level of orlop deck. This arrangement, however, left the outboard and after parts of the engine room without ventilation. This, no doubt, will be remedied in future designs by ventilating blowers delivering fresh air into the dead-air spaces.

COMPARISON.

A comparison of the daily runs showed that the *Virginian* made somewhat better speed than the *Victorian.* It was not apparent whether the increased speed of the former vessel was due to greater coal consumption or to the more careful construction of the turbines, or entirely to the influence of the wind and sea.

A practical test on each of the vessels showed that the auxiliaries used 11 per cent. to 12 per cent. of the total boiler power. This result was obtained by using one boiler exclusively for running the auxiliaries.

The average speed taken from twelve representative daily runs was 15.3 knots, and to maintain this speed it is estimated from observation of firing that the total daily coal consumption was 172 tons. The exact coal consumption could not be obtained from the company's officials.

Deducting from this total daily coal consumption 12 per cent. for the auxiliaries, leaves 172—20.6=151.4 tons for the *Victorian's* propulsion alone.

There being no direct means of determining the I.H.P. of large marine turbines, it becomes necessary to arrive at their power by comparison with known results from similar vessels. On the trial of the S. S. *Friederich der Grosse*, a vessel similar to the *Victorian* in dimensions, model and speed, it was found that 7,200 I.H.P. gave a speed of 14.8 knots.

FRIEDERICH DER GROSSE.

Length, feet and inches	525-10
Beam, feet	60
Draught, mean, feet	27
Gross tonnage	10,526
Twin screws.	
Engines, quadruple-expansion, inches	$25\frac{1}{4} \times 35\frac{3}{16} \times 52\frac{1}{2} \times 75\frac{5}{8}$
Stroke, inches	$55\frac{1}{4}$
Coal consumption per 24 hours for propulsion, tons	115.5
per I.H.P. per hour for propulsion, pounds	1.54

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Raising the speed of the *Freiderich der Grosse* to 15.3 knots would require

$$\frac{15.3^3}{14.8^3} \times 7{,}200 = 7{,}956 \text{ I.H.P.}$$

Taking the ratio of the gross tonnage of the two ships indicates that the *Victorian* would require

$$\left(\frac{11{,}200}{10{,}526}\right)^{\frac{1}{2}} \times 7{,}956 = 8{,}544 \text{ I.H.P. for a speed of } 15.3 \text{ knots.}$$

With reciprocating engines (based on *Friederich der Grosse's* performances of 1.54 pounds per I.H.P. per hour) the *Victorian* would require per 24 hours for propulsion alone $\frac{8{,}544 \times 1.54 \times 24}{2{,}240} = 141$ tons, in order to get results equal to those obtained on the *Friederich der Grosse.* Attention is called to certain conditions which effect the economical results. The *Friederich der Grosse's* engines were quadruple-expansion and used steam of 213 pounds pressure, while the *Victorian* used steam pressure of only 160 pounds. The vacuum of the port condenser was not good during several days owing to air leak, perhaps through the steam packing of port L.P. turbine. Besides, as mentioned previously, the boilers primed at times, which no doubt interfered with the economic performance of the turbines. The disturbing influence of the above-mentioned conditions no doubt will be overcome entirely or materially reduced, bringing the economic operation up to good reciprocating practice.

As results appear at present the *Victorian* is using 151.4—141=10.4 tons more coal per twenty-four hours than would have been used with good quadruple-expansion engines, using steam 210 to 220 pounds per square inch.

It is anticipated that this excess of coal consumption per day will be reduced even for the *Victorian's* turbines, and will disappear entirely with future designs, when it is more fully understood just how much the propellers ought to be reduced and just how much the speed of the rotation of the turbines should be increased. What has already been done at this early day in the improvement of the turbine presages its speedy development

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for marine purposes. On land, in power plants with turbine motors, results have been obtained showing its economic superiority to the reciprocating engines.

OILING.

The number of bearings and working parts for turbine machinery is reduced to a minimum, and in consequence the oiling arrangement is simplified and the expense for oil reduced below that necessary for reciprocating engines. For the main bearings of the turbine there is but little waste, as the closed lubricating circuit allows the oil to be used over and over.

There is, however, no reduction in the oil expenses for the auxiliary machinery, as it is of the usual type installed with reciprocating engines.

PERSONNEL.

It has been suggested that with the less complicated machinery of the turbines that a considerable reduction would be possible in the engineer's force, but as the force necessary for the auxiliary machinery and for the fire rooms is about 85 per cent. of the total force, and remains approximately the same whether the main engines are turbine or reciprocating, very little reduction in engineer's force is possible.

An inspection of the list of men in the engineer's force of the *Victorian* shows that it cannot be reduced judiciously beyond its present numbers.

ENGINEER'S FORCE ON *VICTORIAN.*

1 Chief engineer.
7 Assistant engineers.
6 Engine-room oilers.
3 Leading firemen.
30 Firemen.
21 Coal passers.
1 Storekeeper.
1 Assistant storekeeper.
1 Electrician.
3 Dynamo oilers.
1 Oiler for refrigerating machinery.
1 Machinist for deck auxiliaries.

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SYNOPSIS OF STEAM LOG.

On Table I is tabulated the observations of pressures and revolutions for the turbines and some of the more important auxiliaries.

Table I.—SYNOPSIS OF STEAM LOG—S. S. *VICTORIAN.*

Data taken on trip Halifax—Liverpool—Montreal.

Boilers.					Turbines.				Condensers.						Pumps.	
	Air pressure, inches water.				Gauge pressures at inlet.				Vacuum in inches.				Temp.			
									Main.		Augmentary.					
Pressure per sq. in. (gauge)	Blowers casing.	Ash pit.	Feed heat pressure per sq. in.	Blowers, revolutions per minute.	H.P.	Stb'd L.P.	Port L.P.	Revolutions per minute.	Stb'd.	Port.	Stb'd.	Port.	Inj.	Disch.	Circ. pump, K. P. M.	Air pump, doub. str.
145	2¾	¾	5	...	100	12	12	210	27	27	27	27	...	...	...	40
155	2¾	...	6	...	100	14	14	220	27½	27½	27½	27½	...	...	...	...
170	3	1	6	275	133	18	18	232	25	25	...	...	48	72	160	40
170	...	...	6	...	140	20	20	236	24	24	24	24	...	...	...	...
165	...	...	6	280	150	21	21	238	27	27	27	27	...	...	...	38
165	...	...	5	...	140	22	22	210	27	23	27	23	...	...	...	...
150	...	...	6	280	120	15	15	*	27½	24	27½	24	...	...	...	26
150	...	...	6	...	120	15	15	†	26	19	25	19	...	...	...	...

* H.P. = 220; starboard L.P. = 220. † Port L.P. = 210; 200.

Table II.—SYNOPSIS OF LOG, S. S. *VICTORIAN.*

	Date, 1905.	Sailed from—	Arrived at—	Latitude.	Longitude.	Distance.	Speed.*
Halifax to Liverpool.	April 11.	Halifax, 6 A. M.	.....	*North.*	*West.*		
	11.	.....	.....	43·50	61·50	89	14.8
	12.	.....	.....	42·28	54·00	357	15.2
	13.	.....	.....	42·23	46·00	370	15.8
	14.	.....	.....	45·53	39·00	365	15.6
	15.	.....	.....	49·02	32·04	342	14.6
	16.	.....	.....	51·50	23·59	353	15.1
	17.	.....	.....	54·24	15.10	356	15.2
	18.	Moville, 1·30 P. M.	Moville, 9·30 A. M.	...	...	...	...
	19.	.....	Liverpool, 2·30 A.M.	...	...	195	15
Liverpool to Montreal.	April 27.	Liverpool, 6 P. M.	.....	...	...	...	...
	28.	.....	Moville, 1·30 P. M.	...	...	195	10
	28.	Moville, 8·30 A. M.	.....	...	...	53	15.2
	30.	.....	.....	54·36	19·30	393	15.9
	May 1.	.....	.....	53·11	29·45	372	15.1
	2.	.....	.....	50·32	39·20	388	15.7
	3.	.....	.....	47·37	46·04	317	13
	4.	.....	.....	45·51	53·15	341	13.9
	5.	.....	.....	47·32	59·05	280	11.5
	6.	.....	.....	48·40	62·26	152	6.2
	7.	.....	.....	47·39	70·03	344	14.1
	7.	.....	Quebec, 7 P. M.	...	...	...	...
	8.	Quebec, 5 A. M.	Montreal.	...	...	...	...

* Average speed in knots per hour.

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It will be noted in the last three readings that the vacuum in the port main condenser was considerably lower than for the starboard condenser. This falling off is attributed to an air leak, as the air and circulating pump of this condenser were run at the same speed as those on starboard side.

DAILY RUNS.

Table II gives the daily runs for both the east- and west-bound voyages. The last column gives the average speed in knots per hour, corrected for the difference of longitude.

During May 3d and 4th a moderate gale from the west and heavy head seas caused a decided falling off in speed. The reduced runs for May 5th and 6th are due to delays caused by ice in Gulf of St. Lawrence.

CONCLUSIONS.

There exists a difference of opinion between those interested as to the final status of the steam turbine. The energy and intelligence now bestowed on the turbine motors will decide the position it is to take with its time-honored competitor, the reciprocating engine.

In new power installations on land the turbine has gained a footing that warrants the belief that it has come to stay, the character of work in large power plants admitting of arrangements that will insure the units operating at all times on a full and constant load. Under these conditions the turbine has proved its superiority over the reciprocating engine. With this advantage in its favor it is, however, not believed that it will displace expensive reciprocating machinery already in place and doing its work to the satisfaction of owners and patrons.

The change from reciprocating engines to steam turbines for marine propulsion is not so easily made, owing to the difficulty met at high speeds of rotation in overcoming economically the inertia of water acted on by the propeller, and, further, the falling off in economy when the speed of rotation of turbines is reduced to suit the advantageous but slower rotary speed of the

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propeller. An equitable compromise between the speed of rotation of the turbine and propeller for maximum power and speed is upset and economy sacrificed when a reduced speed and power is attempted. As a solution, the Parsons Steam Turbine Company fit small cruising turbines in naval vessels for use when cruising speed with consequent reduced power is desired.

The cruising turbines require additional engine-room space and add somewhat to weight of propelling machinery. In the present development of the marine turbine this increase in necessary space and weight is not considered a loss, as the combination of the cruising with the main turbines enables the vessels so fitted to keep the sea longer, while still retaining in reserve the advantage of speed.

The length fore and aft of the *Victorian's* engine room is 39 feet 6 inches, and the three turbines and two main condensers being placed abreast, as shown on Fig. 1, occupy the space from side to side of the vessel. There is no gain in floor space for turbine over that required for reciprocating machinery of equal power, but in head room there is a decided advantage in favor of the former, especially in naval vessels.

At the *Victorian's* representative speed of 15.3 knots per hour obtained with 229 revolutions per minute, the slip of propeller was 20.4 per cent.—a very good result so far as slip is concerned, and indicates that the search must be made in other directions for elements that are to make the steam turbine more nearly what is needed for marine purposes. It has already been mentioned that there was considerable priming, and although this condition is not a menace to the turbine, it acts as a great mechanical handicap. The resistance offered to the passage of water among the vanes instead of steam is considerable and must react equally to resist the rotation of the turbine. From experiments made to determine the action of superheated steam, it was found that the efficiency of the turbine was increased 10 per cent. by using steam superheated 100 degrees Fahrenheit. These results indicate an item of economy that should be taken into account in adapting the

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steam turbine for marine purposes. It is not believed, however, that the use of superheated steam in marine turbines will give as good results as those just mentioned, which were obtained from comparatively small units. The large marine turbines will require additional clearances to allow for the expansion resulting from high temperatures of superheated steam.

The minimum clearances, the proportions for nozzles and openings between vanes, the best speed of rotation for marine purposes, etc., can be best obtained by numerous and carefully-conducted experiments made with turbines installed in vessels.

In Great Britain there are at present twenty-four vessels of various classes, either built or building, fitted with marine turbines, indicating a confidence in this type of marine motor. It is thought that this extensive introduction is due to a more substantial cause than the fascination of novelty. Possibilities have been discerned that outweigh the shortcomings shown by the poor performance of some marine turbines.

Important and interesting results have been gotten from trials of H. M. S. *Amethyst* and *Topaze*, sister ships, one with turbines and the other with reciprocating engines, showing that at a speed of 10 knots the turbines used about 19 per cent. more steam than the reciprocating engines. At 14 knots the steam consumption was the same for each type. At 18 knots the turbines used 20 per cent. less, and at 20 knots about 30 per cent. less steam than the reciprocating engines. These results mark important and flattering progress for the marine turbine, and strongly suggest that its use should be more general in vessels of the U. S. Navy.

It is therefore strongly recommended that hereafter turbines be installed in torpedo boats, destroyers, scout cruisers, and in gunboats. Their use in such vessels and the experiences gained therefrom will, no doubt, in time demonstrate the wisdom of fitting them in battleships and armored cruisers.

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NOTES.

STEAM TRIALS OF H. M. S. *CARNARVON.*

In the following description of the *Carnarvon*, taken from "Engineering," of London, March 17, there will be noted a number of points of more than ordinary interest.

The most striking departures from the usual naval practice will be found in the system of forced lubrication for main engine bearings and the arrangements for the recovery of oil from crank pits, the omission of auxiliary condensers and the conditions under which the trial was run, the latter being such as would exist when in ordinary service.—EDITOR.

The armored cruiser *Carnarvon*, the first of the *Devonshire* class to be completed, has passed through her steam trials, and the results have proved thoroughly satisfactory, the mean speed attained on the eight-hours' full-power trial being 23.3 knots, while only 22.25 knots was anticipated in the design. This result is the more remarkable as it closely approximates to the average performance of the earlier cruisers of the County class, which are by no means so well armored, and have a much less powerful armament. The County cruisers were laid down at a time when there was a desire, while increasing the speed, to decrease the displacement and cost; consequently the tonnage was limited to 9,800 tons, and a speed of 23 knots was aimed at. In order to meet these conditions it was decided to have only 4½-inch armor on the broadside amidships, reduced to 3 inches forward, and ultimately to 2 inches at the ram; the conning tower, however, was of 10-inch armor. The armament consisted of fourteen 6-inch guns. Experience has shown that the limitations imposed were not justified. Even at the time the cruisers were built experts were satisfied that 4½-inch armor was quite inadequate to resist penetration by 6-inch and 7-inch modern high-ballistic guns, and that the building of a ship costing from £750,000 to £800,000, armed only with

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6-inch guns—irrespective of the number of such weapons—did not anticipate to any degree the progress in the power and range of enemy's guns.

When the *Devonshires* were ordered, early in 1902, they were to be of the same type as the County cruisers, so far as the thickness of armor and the number and disposition of the guns were concerned; but, following upon the appointment of Mr. Philip Watts as Director of Naval Construction, the designs were considerably modified. The first step taken was to increase the thickness of the armor, the broadside plating being increased to 6 inches instead of 4½ inches, tapering at the forward end to 4½ inches and at the ram to 2 inches. There was a corresponding increase in the thickness of the bulkhead at the after end of the main belt, which is now 4½ inches, while the conning tower has been made of 14-inch armor. The belt extends from well abaft the engine room to the bow; the depth amidships is practically 11 feet; but forward, owing to the sheer of the main deck and the droop of the protective deck towards the ram, this depth is increased to about 15 feet. This increased protection involved deeper draught; and later a further change was made, to considerably increase the power of the armament, as shown in the table of comparative particulars on another page. The early County cruisers had fourteen 6-inch guns; the *Devonshires* were designed to have two 7.5-inch guns and ten 6-inch guns, the former being bow and stern chasers. But in view of the strong opinion which had developed against the once popular 6-inch gun, because of its limitations in energy and range, it was decided to increase the number of 7.5-inch guns to four and to reduce the 6-inch guns to six. Thus, in the cruisers of the *Devonshire* class there are three 7.5-inch bow-chasing guns, one in a barbette on the forecastle and two others in barbettes on the upper deck at the after end of the forecastle. The bow fire thus becomes approximately fifteen shots of 200 pounds weight, each with a muzzle energy of about 11,600 foot tons, against forty-two shots of 100 pounds weight, each with an energy of from 5,000 foot tons to 5,500 foot tons. In view of the increased energy

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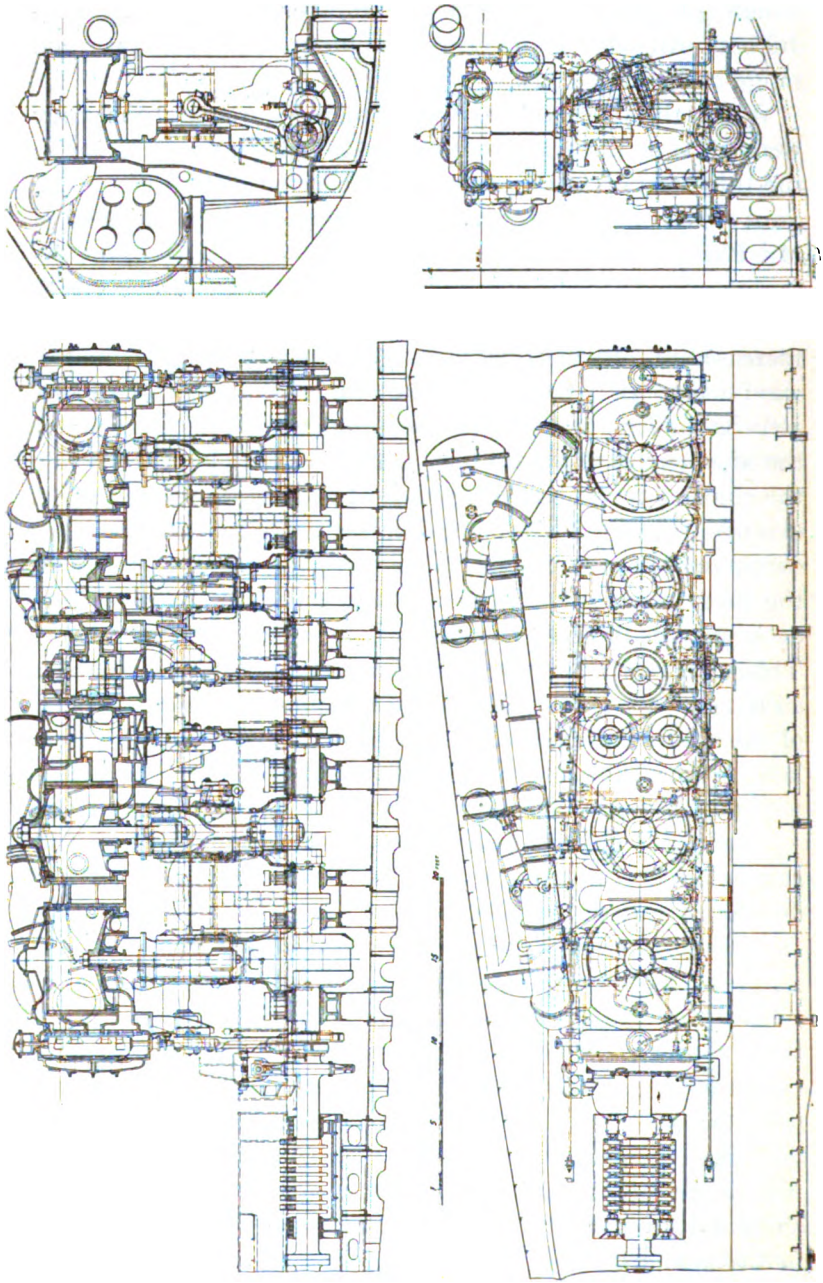

ONE MAIN ENGINE H. M. S. "CARNARVON."

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per shot, no two opinions can be entertained as to the superiority of the bow fire of the *Devonshire* class. As regards the stern fire, there is mounted on the upper deck one 7.5-inch gun in the barbette, while on each side there are double-story casements for 6-inch guns, the lower gun in each pair being on the main-deck level. Amidships on the main deck there is, on each broadside, one 6-inch gun. It seems, however, doubtful whether the 6-inch guns on the main deck can be fought in any seaway, because, apart from the rolling of the ship, there are always waves, so that the projectile may bury itself instead of ricocheting. It would appear, therefore, as if the four 6-inch guns placed on the main deck in this *Devonshire* class should have been dispensed with and additional 7.5-inch guns mounted on the broadside. The smaller guns in the ship include twenty 3-pounder quick-firing guns and two boat and field quick-firing guns of the 12-pounder 8-hundredweight class, in addition to two Maxim guns.

Table I.—COMPARATIVE PARTICULARS OF COUNTY AND *DEVONSHIRE* CLASSES OF CRUISERS.

	County Class.	*Devonshire Class.*
Length between perpendiculars	440 ft.	450 ft.
Breadth (molded).....	66 ft.	68 ft. 6 in.
Depth (molded).....	38 ft. 6 in.	38 ft. 6 in.
Draught (mean).....	24 ft. 6 in.	25 ft.
Displacement (normal)	9,800 tons	10,850 tons
Designed speed.....	23 knots	22¼ knots
Indicated horsepower...	22,000	21,000
Coals (normal).....	800 tons	800 tons
(bunkers full)....	1,750 tons	1,750 tons
Number of boiler rooms and funnels	3	4
Main Armament:	Fourteen 6-in. quick-firing	Four 7.5-in. and six 6-in. quick-firing
On forecastle deck.....	Two 6-in. guns, twin-mountings in barbette	One 7.5-in. gun in barbette
Upper deck at aft end of forecastle	Two 6-in. guns, one on each broadside, in casemates	Two 7-5-in. guns, one on each broadside, in barbettes
Upper deck aft under shelter	Two 6-in guns, one on each broadside, in casemates	Two 6-in. guns, one on each broadside, in casemates

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	County Class.	*Devonshire Class.*
Upper deck aft of shelter	Two 6-in. guns, twin-mountings in barbette	One 7.5-in. gun in barbette
Main deck forward.....	Two 6-in. guns, one on each broadside, in casemates immediately under upper-deck guns	None
Main deck amidships...	Two 6-in. guns, one on each broadside, in casemates	Two 6-in guns, one on each broadside, in casemates
Main deck aft.....	Two 6-in. guns, one on each broadside, in casemates immediately under upper-deck guns	Two 6-in. guns on each broadside, in casemates immediately under upper-deck guns
Smaller guns:		
12-pounder 12-cwt. quick-firing	8	None
12-pounder 8-cwt. boat and field quick-firing	2	2
3-pounder quick-firing	3	20
0.303 Maxim automatic	2	2

The alterations made in the various ships have considerably delayed them. Messrs. William Beardmore and Co. are the first to complete one of the vessels of her class, an honor which also fell to them in connection with the completion of the *Berwick* of the County class. The next vessel to go on trial will be the *Antrim*, built and engined by Messrs. John Brown & Co., of Clydebank, to be followed by the *Devonshire*, built at Chatam Dockyard; the *Hampshire*, constructed by Sir W. G. Armstrong, Whitworth & Co., Newcastle-on-Tyne; the *Roxburgh*, by the London and Glasgow Company; and the *Argyll*, by Scott's Shipbuilding and Engineering Company, of Greenock.

In respect to boiler installation, there was also considerable range, as the plans for these ships were submitted to the Boiler Committee, and it was decided to adopt various systems of boilers. In all cases part of the installation consists of cylindrical boilers, as first recommended by the Boiler Committee; but experience has since shown that this is not a de-

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sirable procedure. Not only does it involve complications, but it has been proved that the efficiency and economy of the cylindrical boiler are not equal to the results achieved by various types of water-tube boilers, working under exactly corresponding conditions in the same ship. The *Carnarvon* has Niclausse boilers, and cylindrical boilers with the closed stokehold system of forced draft; the *Devonshire* has Niclausse and cylindrical boilers on the Howden system; the *Roxburgh* has the Dürr boiler, with closed-stokehold forced-draft for the cylindrical boilers; the *Antrim* has Yarrow and cylindrical boilers, both forced on the Howden system; while the same boilers in the *Hampshire* are on the closed-stokehold system. The *Argyll* has closed stokeholds for the cylindrical boilers, but natural draft is adopted for the Babcock & Wilcox boilers. It remains to be seen whether full advantage will be taken of these variations for comparative experimental tests. Time, a staff, and money must be found, if the confusion involved in the variety is to be justified.

Confining ourselves now to the *Carnarvon*, the machinery for which has been constructed by Messrs. Humphrys, Tennant & Co., of Deptford, London. It will be noted that there are six single-ended tank boilers in the aft stokehold, while in each of Nos. 2 and 3 stokeholds there are eight boilers of the Niclausse type; in the forward stokehold, where the lines of the ship reduce the width of the hold, there are only six. In each stokehold there are fitted two of Weir's feed pumps and a See ash ejector. There are thus twenty-two Niclausse water-tube and six cylindrical boilers. Fourteen of the Niclausse boilers have 15 sections, six have 14 sections, and two 12 sections, each section consisting of a header with twenty-two steel tubes, 3.3 inches in external diameter; the headers are connected to the steam collectors by coned joints. The total heating surface in the twenty-two Niclausse boilers is 43,566 square feet, and the grate surface 1,224 square feet, the length of firebars being 6 feet 6½ inches. The cylindrical boilers are 13 feet in diameter and 9 feet 1 inch long, each having three corrugated furnaces 3 feet in inside diameter, and

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364 steel tubes 2½ inches in external diameter, and 6 feet 0½ inch long between the tube plates; the total heating surface in the six boilers is 10,158 square feet, and the grate surface 324 square feet, the length of the firebars being 5 feet 9½ inches.

The propelling machinery consists of two sets of four-cylinder triple-expansion engines arranged in separate water tight compartments, with a longitudinal bulkhead. The engines were designed to develop collectively 21,000 indicated horse-power at 138 revolutions per minute, with a pressure in the boilers of 210 pounds per square inch. The cylinder covers, pistons and bottom frames are of cast steel. The cylinders of each set of engines are supported at the front by six forged-steel turned columns, and at the back by four cast-iron rectangular columns, on which are bolted the guides for the piston-rod shoe plates. The diameters of the cylinders are high-pressure, 41½ inches; intermediate, 65½ inches; and two low-pressure, 73½ inches; all having a stroke of 42 inches. All the cylinders are fitted with liners, and are steam jacketed. The high-pressure and intermediate-pressure liners are made of solid forged steel, and those for the low-pressure cylinders of cast iron. The distribution valve on the high-pressure cylinder is of the piston type; the intermediate cylinder has two piston valves, operated by seperate spindles connected to a crosshead. Slide valves are provided for the two low-pressure cylinders. Starting valves are fitted to the intermediate and low-pressure valve casings. The reversing gear is of the well-known Humphrys type, reversing being effected by means of a screw; there is also an automatic cut-off to the main engine.

The shafting is hollow throughout; the crank shaft is 18 inches in diameter, with a 9-inch hole; the crank pins are all 20 inches in diameter, those for the high-pressure and intermediate-pressure being 26 inches long, and for the low-pressure 17 inches long. The angles of the cranks are arranged on the Yarrow, Schlick and Tweedy system, to balance the weight of the moving parts. Forced lubrication has been applied experimentally to some of the bearings in the *Carnar-*

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von, the intention of the Admiralty being to develop a satisfactory system in connection with the whole of the crank-shaft bearings, the main bearing, the crank heads and eccentrics. Messrs. Humphrys, Tennant & Co. have carried out extensive experiments on a large scale at their works, and they have fitted one of the main bearings and one of the crank heads of the *Carnarvon* with a system of forced lubrication, which, having proved very satisfactory, is now to be applied to all the bearings in the battleship *Britannia*, the engines of which are now being made by the same firm. The oil is pumped into the main bearing, in which it is confined by a species of gland on each end of the bearing; thence it is conveyed by means of holes to the center of the crank shaft, and in the usual manner to the crank-pin, and finally into the crank brasses. The oil is recovered from the crank pit, passed through filters, and again pumped through the main bearing in the same cycle. To reduce to a minimum the water required, special glands are fitted to the piston rod, so that any water passing the main gland can pass away to the bilge. In orher vessels of the *Britannia* class a different system is to be experimented with, in order to obtain experience and to evolve finally the most effective method. In connection with this, Messrs. John Brown & Co., who are building the machinery for the *Africa*, are also carrying out a series of important experiments. The Admiralty are indebted to such firms for devoting their experience and resources for the advancement of engineering in the service.

It will be noticed that the system of lubrication of the thrust block in the *Carnarvon* differs from that which we have hitherto seen in connection with naval machinery. The lubrication is in this case entirely effected by means of grooves cut on the collars of the thrust shaft, which in rotation dip into the oil contained in the well formed in lower part of the block. The oil is retained in this by means of small casings fitted between each separate collar. The temperature in the block itself is, therefore, a fair guide to satisfactory working. The system worked well during the trial of the *Carnarvon*, at a tempera-

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ture which, judged by immersion of the finger point, seemed much greater than that possible under ordinary water cooling. The rings are solid; but we understand that with the next application of the same system water-cooled collars will be adopted, as experiments at the works of Messrs. Humphrys, Tennant & Co. have shown that a remarkable lowering of the temperature of the oil in the thrust block is effected when the heat generated is conveyed away by means of water passing through the collars.

There are two condensers for each engine. The one centrifugal pump for both condensers is placed on the upper platform, while on the lower platform there is located a filter tank. The condensers have each 6,575 square feet of cooling surface, and either may be shut off for overhaul, as the other is sufficient. Independent twin air pumps of the Weir type are placed forward of the condenser plant. They draw from the condenser, and discharge through grease filters into the feed tank. There is no auxiliary condenser in the ship, the exhaust from the auxiliary machinery being passed into the main condensers, or into the closed exhaust system, or into the receiver of the low-pressure cylinder of the main engine, or into the evaporator. There is one distiller, located at the after end of each engine room. This distiller is capable of taking the total output of both evaporators for making up the boiler feed. In many previous ships the boiler make-up was provided by one evaporator discharging into the main condenser; but in this case the evaporating and distilling system is a distinct unit of itself. The advantage of this is that the distiller, taking the whole of the output of both evaporators, gives feed water at a much higher temperature than is the case when it is working in conjunction with one evaporator for the production of drinking water. The rate of output of fresh water is adjusted in each case, to insure a high or low temperature of fresh water at the outlet. The engine rooms are identical in arrangement, and have donkey and bilge pumps.

The trials of the *Carnarvon* were carried out under the new service conditions, which require that the staff employed and

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the arrangements on board shall agree exactly with those obtaining when the ship is in war service. Thus all bulkhead doors were shut, the engine room was closed down, the staff was restricted to the service complement, the amount of oil used was limited, and no water was permitted on the bearings. Clearances were also prescribed. In view of the severity of these conditions when applied to new machinery, the proportion of power required to be developed on the second thirty-hours' trial was reduced from 80 per cent., as in previous cruisers, to 70 per cent. of the full power. This is the power which can be maintained as long as the coal may last and can be brought to the stokehold plates. No difficulty whatever was experienced in obtaining the results, and as regards the temperature of the closed-down engine room, the highest reading—and that for a short period only—was 90 degrees Fahrenheit, while in the ammunition passages and other parts of the interior it was 75 degrees Fahrenheit. The outside temperature was, if anything, higher than is usual at this period of the year. The ease with which ship after ship is passing through her trials under the new conditions clearly shows that no trouble need be anticipated, even in running for thirty hours at 80 per cent. power, and in the future the sea-going engineer will not be able to point to any favorable circumstance during trials in extenuation of any inability to maintain the legend speed in service, an excuse sometimes, although not often, made.

On the first trial, which was of thirty hours' duration, the vessel steamed from the Clyde to Portsmouth, and experienced very severe weather, the wind blowing with a force of 10 for the greater part of the time. The steaming results are set out in Table II, from which it will be seen that the mean power was 4,756 horsepower, while the coal consumption worked out to 2.11 pounds per indicated horsepower, and the loss of water to 2.14 tons per 1,000 indicated horsepower per twenty-four hours. Before returning to the Clyde the vessel proceeded to the English Channel to carry out her gun trials, with a view of testing the mountings and adjacent ship structure. The

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Table II.—STEAM TRIALS OF H. M. S. *CARNARVON.*

Duration of trial, hours	30		30		30	
Nature of trial	One-fifth power coal consumption.		70 per cent. power coal consumption.		Full power.	
	Starboard.	*Port.*	*Starboard.*	*Port.*	*Starboard.*	*Port.*
Mean steam in boilers, pounds	184		205		202	
at engines (H.P. receiver), pounds	89	97	171	170	173	173
air-pressure in stokeholds,* inch	...	...	0.4	0.4	0.7	
cut-off in H.P. cylinder, per cent	...	...	59.8	59.8	77	77
vacuum, inches	26.1	26.9	26.8	27.3	26	25.5
revolutions	84.2	84.4	129.2	128.3	141.7	142.5
Mean pressure in cylinders, high, pounds	36.8	39.8	73.8	73.9	89.5	90.9
intermediate, pounds	10.6	11.72	24.4	25	29.7	30.1
low, forward, pounds	5.44	5.5	12.35	11.84	15.45	16.66
aft, pounds	4.64	5	10.9	10.2	15.12	15.79
I.H.P., high	889	963	2,692	2,722	3,640	3,727
intermediate	638	707	2,250	2,295	3,008	3,064
low, forward	411	417	1,436	1,367	1,971	2,138
aft	351	380	1,264	1,186	1,929	2,012
Total I.H.P.	2,289	2,467	7,642	7,570	10,548	10,941
Gross total I.H.P.	4,756		15,212		21,489	
Coal consumption per I.H.P., pounds	2.11		1.78		2.29	
Speed of ship, knots	14.3		21.43		23.3	
Total loss of water, tons	12.76		44.02		25.5	
per 1,000 I.H.P. per 24 hours, tons	2.14		2.31		3.5	
Water made by evaporators, tons	12.74		41.07		...	

*Only applicable to stokehold with cylindrical boilers.

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gun mountings were fitted by Sir W. G. Armstrong, Whitworth & Co., and gave every satisfaction, while the ship withstood the stresses splendidly.

The second trial of thirty hours' duration was carried out when the vessel was returning to the Clyde. This trial took place on Tuesday and Wednesday of last week, and again very severe weather conditions were met with, the wind being of force 8 to 10. The results are given in Table II. There was an air pressure of 0.4 inch in the stokehold with cylindrical boilers. The power developed was 15,212 indicated horsepower, the coal consumption being 1.78 pounds per horsepower hour, and loss of water 2.31 tons per 1,000 indicated horsepower per twenty-four hours. The contract required that during this run the power should be 14,700, which, it will be noted, was exceeded by 500 horsepower. During this trial, on the 8th instant, two runs were made on the measured mile at Skelmorlie on the Clyde. On the first run the engines made 132.8 revolutions, and developed 15,304 indicated horsepower, which gave a speed of 22.195 knots. On the second run the mean revolutions were 129.6, the power developed being 15,398, while the speed was 20.666 knots. The mean results were: Revolutions, 131.2; power, 15,351 indicated horsepower; speed, 21.43 knots. The displacement when the vessel was running the mile was 10,500 tons. The later cruisers of the County class made on this trial a speed of about 22 knots, with about 16,200 indicated horsepower, so that the *Carnarvon* has done exceedingly well, because not only is the power and aggregate coal consumption less, but the displacement was 1,700 tons greater. The increased armor and armament has thus been bought for a small forfeiture in speed—about half-a-mile per hour with the same power as in the County class.

The full-power trial took place on Friday, March 10th, when again the weather was unpropitious, a wind force of 4 to 6 blowing, with frequent squalls, of long duration, of force 10. On this trial the vessel started a little in excess of her designed displacement, and when four hours had elapsed she was put on the measured mile at Skelmorlie. It was then slack tide, so

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that the only interference was due to the wind. It will be noted from Table III that the first and third runs occupied exactly the same time, which was also the case with the second and fourth runs, but the power varied. The fact that the speed is the same, notwithstanding the difference in power, is, of course, due to the difference in wind resistance. The mean speed was 23.3 knots, the engines having made 142.3 revolutions, and the indicated was 21,322 horsepower. The highest speed got with any of the vessels of the County class was 24 knots, with 22,881 indicated horsepower; and here again the performance of the *Carnarvon* must be regarded as satisfactory, for the loss in speed is only 0.7 mile per hour, notwithstanding that the power is 1,500 indicated horsepower less and that the displacement is 1,000 tons more. A high propulsive efficiency seems to have been attained, which is due partly to increased length and to the propellers. These are of Stone's manganese-bronze, 15 feet 9 inches in diameter, each having three blades, set at a pitch of 19 feet 6 inches. The total blade surface in each propeller is 80 square feet.

Table III.—FOUR RUNS ON MEASURED MILE AT FULL POWER.

Number of run.	Time.	Revolutions per minute.		Indicated horsepower.			Speed, knots.
		Starboard.	Port.	Starboard.	Port.	Total.	
	m. s.						
1	2 35.2	143.8	144.6	10,770	11,046	21,816	23.196
2	2 33.8	142.4	143.5	10,408	10,959	21,367	23.407
3	2 35.2	141.9	142.6	10,305	10,798	21,103	23.196
4	2 33.8	141.6	142.8	10,333	10,669	21,002	23.407
Mean of means	...	142.3	143.2	10,454	10,868	21,322	23.3

The water-tube boilers, which, as already indicated, are of the Niclausse type, as fitted and proved satisfactory on board the *Berwick* and *Suffolk*, gave plenty of steam and involved no trouble, the automatic feed working very reliably. The mean results of the eight-hours' trial are recorded in Table II, and it will be seen that the steam pressure at the boilers was 202 pounds and at the engines 173 pounds. The cylindrical

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boilers required an air pressure of 0.7 inches, and even then did not maintain their fair proportion of steam at full pressure. The vacuum was 25.75 inches, and the average power for the eight hours was 21,489 indicated horsepower. The coal consumption worked out to 2.29 pounds per indicated horsepower per hour.

After the steam tests were over, the vessel was put through a number of maneuvers, and the steering gear (fitted by Messrs. Napier Brothers, limited, of Glasgow) was tried. There are two sets of vertical engines located in the engine room. They operate through shafting a screw with a right- and left-handed thread, on which are nuts connected to the crosshead on the rudder head. Special note was taken of the smooth running of the steering engines and of the facility with which the one was uncoupled and the other put in gear after the orders were given. One of the engines ran the rudder over from the center line to hard-a-port in 10 seconds, while the other engine ran the rudder over from hard-a-port to hard-a-starboard in 18 seconds, both very satisfactory results. Circling trials were conducted and other maneuvers and tests made.

THE STEAM TRIALS OF H. M. SS. *ANTRIM* AND *DEVONSHIRE*.

Two more of the armored cruisers of the *Devonshire* class have now passed through their trials, attaining in each instance very satisfactory results, and demonstrating that, although the legend speed was only 22¼ knots, all of the vessels will attain 23 knots, largely as the result of recent experience regarding the proportions of propellers. The first vessel tried—the *Carnarvon*, built by Messrs. William Beardmore & Co., Limited, Dalmuir, (see p. 430) attained a speed of 23.3 knots with 21,322 indicated horsepower. The *Antrim*, built and engined by Messrs. John Brown & Co., Limited, of Clydebank, steamed at 23.02 knots for 21,606 indicated horsepower, and the *Devonshire*, built at Chatham, and engined by the Thames Iron Works Shipbuilding and Engineering Company, Limited, made 22.97 knots for just over 21,000 indicated horsepower.

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In these vessels comparative tests are being made as to the efficiency of propellers, and for this reason the speed results are interesting. We gave in our record of the *Carnarvon* trials (page 442) the details of the measured-mile runs, and now we give in Table I the corresponding records of the measured-mile runs by the *Antrim*. In practically all the vessels of the class the propellers are three-bladed, with a diameter of 15 feet 9 inches, and a normal pitch of 19 feet 6 inches. There is, however, a difference in the expanded surface. The *Carnarvon*, for instance, has a total blade surface of 80 square feet, whereas in the case of the *Antrim* it is 75 square feet. Probably, as a direct result of this, the speed of the *Antrim*, as shown in Table I, was 21.33 knots, when the engines were developing 14,643 indicated horsepower, whereas in the case of the *Carnarvon* it was 21.43 knots for 15,351 indicated horsepower; so that the assumption is that at this speed the larger surface propeller was not quite so efficient. At full power, however, although the *Antrim* indicated about 300 horsepower in excess of the power developed by the *Carnarvon*, her speed was considerably lower. The *Carnarvon*, which has proved herself thus far the fastest cruiser of the class, made

Table I.—RESULTS OF MEASURED-MILE SPEED TRIALS OF H. M. S. *ANTRIM*.

Number of runs.	Revolutions per minute.		Indicated horse-power.		Total.	Speed, knots.
	Star-board.	Port.	Star-board.	Port.		
Four Runs on 14,000 Indicated-Horsepower Thirty-Hours' Trial.						
1	125.6	129.6	7,097	7,591	14,688	20.88
2	128.3	129.3	7,233	7,173	14,406	21.92
3	129.6	129.3	7,773	7,178	14,951	20.69
4	131	129.6	7,533	6,995	14,528	21.94
Mean of means.....	128.7	129.4	.....	.....	14,643	21.33
Four Runs on Full-Power Eight-Hours' Trial.						
1	143.3	143	10,698	11,180	21,878	23.31
2	143.3	143	10,664	10,914	21,578	22.99
3	145.3	142	10,928	10,903	21,831	22.88
4	142.3	142.6	10,419	10,718	21,137	23.23
Mean of means.....	143.8	142.6	.....	.....	21,606	23.02

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23.3 knots for 21,322 indicated horsepower, while the *Antrim* for 21,606 indicated horsepower steamed at 23.02 knots.

Another interesting fact in connection with the class is, as we pointed out in connection with the *Carnarvon*, the variation in the types of boilers used. In the first place, a considerable portion of the steam-generating plant is of the cylindrical boiler type; and, in view of the remark made by one of the speakers during a discussion at the recent meetings of the Institution of Naval Architects, it may be noted that comparative trials, in one of the ships now fitted with a proportion of cylindrical and water-tube boilers, have shown that this arrangement, proposed by the Boiler Committee, and now happily departed from, has involved a material reduction in the speed of some of our ships. One of the *King Edward VII* battleships, which has the combination system, was ordered to run four trials of eight hours' duration in order to test the comparative evaporation of the same proportion of cylindrical and Babcock & Wilcox boilers. The first trial was with Babcock & Wilcox boilers, equal to one-fifth total power, and the second with cylindrical boilers, also equal to one-fifth of the total power. It was found that whereas the former gave 3,759 indicated horsepower, the tank boilers only gave 3,634 indicated horsepower; the coal consumption was 1.74 pounds and 1.8 pounds per indicated horsepower per hour respectively. The third trial was with all the cylindrical boilers working under full-power trial conditions, under an air pressure not exceeding 1 inch; while the fourth trial was with an equal proportion of boilers of the Babcock & Wilcox type, burning an equal amount of coal to that used by the cylindrical boilers in the third trial, and run so as to attain as high a power as possible. The cylindrical boilers were supposed, when the design was prepared, to be equal to two-fifths of the full power—namely, 7,200 indicated horsepower; but on this special trial, when worked at an air pressure of 0.96 inch, it was found that the total power got was only 6,686 indicated horsepower, showing that they were deficient to the extent of nearly 10 per cent. The total coal burned on the eight hours' trial

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was 100,890 pounds, equal to 1.88 pounds per indicated horsepower per hour. An equal proportion of Babcock & Wilcox boilers, working at the same power, and burning in the eight hours 100,498 pounds of coal, the air pressure in the stokehold being 0.8, gave 7,510 indicated horsepower, which was 300 more than was anticipated, while their coal consumption was only 1.67 pounds per indicated horsepower per hour. In other words, for the same coal consumption, the water-tube boilers gave over 12 per cent. more power, and the ratio of consumption to power was 0.21 pound less per unit of power per hour = 11 per cent. less than with the tank boilers.

The boiler installation of the *Antrim* is a combination of the Yarrow type and cylindrical boilers. In the forward boiler room there are five, and in each of the two middle boiler rooms six, Yarrow type boilers, making in all seventeen Yarrow type boilers, having a grate area of 869 square feet and a heating surface of 49,077 square feet. The after boiler room contains six cylindrical boilers, having a grate area of 313 square feet and a heating surface of 9,366 square feet. Both the Yarrow type and the cylindrical boilers are arranged to work with Howden's system of forced draft. Two Weir's feed pumps are fitted in each boiler room. In each of the three forward stokeholds there is a See's ash ejector, and in the after stokehold an electrically-driven ash hoist. The fans for the forced-draft system are twenty in number, and are driven by by enclosed steam engines worked with forced lubrication. The air-pressure and coal-consumption results of the *Antrim* trials are recorded in Table II, along with other details. In the high-power runs the evaporation per pound of coal was over 8 pounds. From first to last everything worked most satisfactorily, alike as regards the boilers and their machinery. On the full-power trials the mean power was 21,604 indicated horsepower, which was 604 indicated horsepower in excess of the guarantee; while on the thirty hours' trial, when, according to contract, the power was to average 14,000 indicated horsepower, the mean attained was 14,628 indicated horsepower, with a coal consumption of 1.95 pounds per indicated horsepower per

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Table II.—STEAM TRIALS OF H. M. S. *ANTRIM*.

Duration of trial	30 hours.		30 hours.		8 hours.	
Nature of trial	One-fifth power coal consumption.		14,000 indicated horsepower.		Full power.	
	Forward.	*Aft.*	*Forward.*	*Aft.*	*Forward.*	*Aft.*
Draught of water, feet and inches	24– 8	25– 7½	24– 7½	25– 6¾	24– 7	25– 7
Steam pressure in boilers, pounds	161		186		185	
			Trunks.	*Ashpits.*	*Trunks.*	*Ashpits.*
Air pressure in stokeholds, cylindrical boilers, inches	0.2		0.98	0.65	1.77	0.75
water-tube boilers, inches			0.9	0.4	1.66	0.78
	Starboard.	*Port.*	*Starboard.*	*Port.*	*Starboard*	*Port.*
Vacuum, inches	25.4	25.5	25.5	25.1	25	25
Revolutions	88.4	87.7	127.8	127.4	142.6	142.6
Mean pressure in receivers, high, pounds	104.1	103.6	158.3	159	163.7	166
intermediate, pounds	12.6	13.5	46.5	48.1	52.7	55
low, pounds	10.2	6.18	8.5	6.75	14	15
cylinders, high, pounds	39.22	38.12	67.4	64.5	81.15	82.6
intermediate, pounds	11.99	12.05	26.6	25.4	30.75	32.5
low, forward, pounds	3.79	3.77	11	11	15.8	16.7
aft, pounds	3.70	3.84	10.9	10.8	16.7	16.64
Mean indicated horsepower, high	995	960	2,472	2,358	3,321	3,383
intermediate	758	756	2,426	2,325	3,135	3,315
low, forward	302	299	1,271	1,272	2,026	2,143
aft	295	303	1,258	1,246	2,145	2,136
Total indicated horsepower	2,350	2,318	7,427	7,201	10,627	10,977
Grand total	4,668		14,628		21,604	
Coal consumption per I.H.P. per hour, pounds	2.06		1.95		2.26	
Water consumption (including auxiliary engines) per hour, pounds	18.451		16.48		18.25	
Loss of water per 1,000 I.H.P. per 24 hours, tons	2.2		1.7		2.5	
Total loss of water, tons	12.86		31.75		18	
Speed of ship, average throughout trial, knots	.....		21.1		22.92	
mean of four runs on measured mile, knots	.....		21.33		23.02	

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Table III.—RESULTS OF STEAM TRIALS OF H. M. ARMORED CRUISER *DEVONSHIRE*.

Date of trial	March 13 and 14, 1905.		March 22 and 23, 1905.		April 7, 1905.	
Nature of trial	30 hours' coal and water consumption at 4,500 indicated horsepower.		30 hours' coal consumption at 14,700 indicated horsepower.		8 hours' full-power trial.	
	Forward.	*Aft.*	*Forward.*	*Aft.*	*Forward.*	*Aft.*
Draught of water, feet and inches	24- 6	25- 6	24- 6	25- 6	24- 6	25- 6
Speed, knots	13.07 by bearings.		21 by log.		22.97 mean of four runs.	
Steam pressure in boilers, pounds per square inch	159		193		200	
Air pressure in stokeholds, inch	.....		.....		·595	
	Starboard.	*Port.*	*Starboard.*	*Port.*	*Starboard.*	*Port.*
Vrcuum in condensers, inches	25.2	23.8	25.6	25.9	24.6	25.3
Revolutions per minute	83.8	83.4	126.7	126.8	139.55	140.45
Mean pressure in receivers, high, pounds	110	101	165	161	179	179
intermediate, pounds	21	20	45	45	64	65
low, pounds	3	2	8	9	20	19
Mean pressure in cylinders, high, pounds	33.6	36	70	70.4	85	81.8
intermediate, pounds	13.2	13.05	24.8	26.2	30.9	31.9
low, fore, pounds	4.6	4.07	10.3	10.7	15.4	15.5
aft, pounds	4.53	4.02	11.5	12.1	17.95	18.5
Mean indicated horsepower, high	810	862	2,548	2,564	3,405	3,295
intermediate	787	778	2,249	2,374	3,081	3,200
low, fore	347	305	1,176	1,225	1,935	1,962
aft	342	302	1,310	1,384	2,255	2,342
Total indicated horsepower	2,286	2,247	7,283	7,547	10,676	10,799
Grand total	4,533		14,830		21,475	
Coal consumption per I.H.P. per hour, pounds	2.05		2.023		1.79	
Total loss of water during trial, tons	18.1		52.64		18.82	

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hour, and a water consumption of 16.48 pounds. In the ships with the Yarrow type boilers it was decided that the power on this thirty-hours' trial should be 14,000 indicated horsepower, against 14,700 in the ships with other types of boiler.

In the *Devonshire*, the prototype of the class, the boilers are of the Niclausse type—fully dealt with in connection with the *Carnarvon*. After the second thirty-hours' trial, however, the boilers of the *Devonshire* were modified, baffles being fitted in the steam drums, which effected a marked improvement, as is indicated by the reduced coal consumption on the full-power trial, and also by the reduction in loss of water during the trial. The vessel was designed to develop 25,000 horsepower; but, as is shown, she maintained during the eight-hours' run a mean of 21,475 horsepower. The four runs over the measured mile were made at about the normal power, and the speed was 22.97 knots. The results on all three trials are tabulated in Table III.

As in our article (see p. 430) we fully described the main characteristics of design of ship and engines, it is not necessary here to do more than give the leading dimensions. The length between perpendiculars is 450 feet; the breadth, molded, 68 feet 6 inches; the depth, molded, 38 feet 6 inches, and the mean draught 25 feet, at which the displacement is 10,850 tons. The vessels are fitted with four 7.5-inch and six 6-inch quick-firing guns, along with twenty-five smaller weapons. The engines are of the triple-expansion four-cylinder type, now almost universally adopted in cruiser work, and the diameters of the cylinders are: High pressure, 41½ inches; intermediate, 65½ inches; and two low-pressure, 73½ inches; all having a stroke of 42 inches.

The trials of the *Antrim* and *Devonshire*, as of the *Carnarvon*, were carried out under the new regulations, which require that the staff employed and the general arrangement agree exactly with the service conditions. All bulkhead doors were shut, and the engines were run with limited clearance in the bearings, and without water. The running of the machinery and the general results in all cases were highly satisfactory.—"Engineering," London, April 28, 1905.

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A CHANGE IN BATTLESHIP CONSTRUCTION.

By G. E. R. (From "Ueberall," No. 15, April 7, 1905.)

* * * I have previously spoken of the tendency of the intermediate battery towards considerably larger calibers, and derived the conclusion that the new German battleships also should have a larger medium-caliber gun, while retaining for the present the four 28-cm. guns of the main battery. But in the meantime a view which actually urges the adoption of the 30.5-cm. caliber for the main battery has been gaining more and more ground in England and appears to have prevailed. No doubt a material change in the English battleship type is impending, one indication in that direction is the circumstance that nothing has become public as regards the names and data for the new ships which are to follow the *Lord Nelson* class.

The *King Edward* class as well as the two ships of the *Lord Nelson* class are the results of a period of groping and experimenting in English battleship construction. Especially the ships of the *King Edward* class, with their three different medium and large calbers, are by no means ideal from the point of view of armament, and we may say without hesitation that their fighting efficiency is very low as compared with their displacement. Compare them, for instance, with the very much smaller ships of the German *Braunschweig* class. That does, of course, not alter the fact that, taken absolutely, the *King Edward* is considerably superior to the *Braunschweig*. Even in England many unfavorable opinions may be heard concerning the *King Edward* class as also the *Lord Nelson* class, especially in technical circles, while the press, of course, as usual and in accordance with its principles, lauds and exaggerates the advantages of the English ships over those of other nations.

Several months ago the head of the well-known shipbuilding firm of Vickers made a speech—if I remember right, on the occasion of a launching—in which he referred to the battleship of the future. He said, among other things, that in his

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opinion the ideal type of battleship was one with a battery of ten 30.5-cm. guns and a speed of over 20 knots. The armor protection would, of course, have to be accordingly. In the course of the last few months several superior officers of the English Navy, as also the former First Lord of the Admiralty, Lord Selborne, and the Civil Lord, Lord Lee, have all expressed the opinion that England would in future build only battleships which it would be impossible to surpass in power of armament. Combining all these indications, it may be pretty safely assumed that a battleship having 30.5-cm. guns exclusively is being seriously considered in England, and that the plans for the same have been probably fully worked out. From the point of view of armament there can be nothing beyond that, since it has so far been the general opinion that the 30.4-cm. caliber is the largest that can be used to good advantage on board of warships.

The chief objections to such a uniform armament may be summed up as follows: The rapidity of fire is lower under all circumstances than that of the smaller calibers; a broadside, or firing all of the turret guns, at the same time, is therefore, to a certain extent, precluded. Speaking purely from a military standpoint, each shot from a large gun represents a comparatively greater event and a failure to hit a greater mishap than in the case of smaller calibers. A large gun can stand at best one hundred rounds; after that it becomes useless. Moreover, the ammunition for such a gun takes up so much space and weight that the allowance per gun can be but small. It follows that the whole battery, whether for the first reason or for the last, can function only for a shorter space of time than the strong medium-caliber battery. Hence, for a ship armed exclusively with large guns it is imperative that she should accomplish her military mission—that of disabling her adversary—before the technical exhaustion of her firing efficiency. And this in its turn entails the principle, both strategically and tactically, of offensive carried to the utmost. An undecisive action, which does not disable the eneny will no

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longer be admissible if the 30-cm. gun ships wish to keep within the limits of the efficiency of their guns.

A very forcible illustration on this point is furnished by the recent exchange of the 30.5-cm. guns on board the English *Majestic* class, mentioned on page 279 of the last number of this publication. These ships have completely worn out their guns in the course of a few years, merely by target practice. It may be assumed under all circumstances that, no matter when a war may break out, nearly all the ships of the fleet will go into action with their heavy guns more or less impaired, for target practice with full-caliber ammunition can not be restricted, not even in the case of the heavy guns, especially in view of recent experiences, which have shown that battles at long range have very good prospects of success. To render such long-range battles decisive—using the term to designate ranges beyond the zone of danger from torpedoes—is the very object striven for in arming battleships exclusively, or almost exclusively, with large guns. No doubt such ships will also be very heavily armored, and so the other nations will be compelled to follow in the same footsteps as regards their new types of battleships. If they do not, the English battleships would be able in a short time, by powerful offensive, to bring about a decision in their favor. Their guns have higher perforative power and higher explosive power. They can fire effectively at the more lightly armed and less powerfully protected hostile ships at ranges where the latter will be unable, with their medium-caliber guns, even those of 24 cm., to inflict any injury upon them. Within a very large zone the present intermediate battery would thus be wholly ineffective, and there would remain only the main battery, consisting on board the German battleships of four 28-cm. guns. These would be opposed by ten 30.5-cm. guns, and the issue would not be doubtful, even if two German ships were confronting one English ship. Hence we shall have to follow suit.

It is ardently to be hoped that the new battleships to be asked for in the fall of next year will be armed with eight to ten 28-cm. guns in place of four. There is no reason why we

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should abandon this caliber and substitute a larger one, since the difference in efficiency between the 30.5-cm. and the 28-cm. gun is comparatively small, while, on the other hand, there would probably be a very great saving in weight, which, with equal displacements, could be utilized in favor of the ammunition supply, the armor, etc. Whether it is a good plan to renounce entirely the intermediate battery, as the English have done in the *Lord Nelson* class (in this case the armament consists of four 30.5-cm. and ten 23.4-cm. guns), is open to discussion. While the medium-caliber guns—that is, up to 20-cm.—can, under present and future conditions, no longer claim to have armor-piercing effects, modern ships have now, and will have in future, so many unprotected parts, especially as regards the superstructures, that it would seem risky to relinquish the fire of explosive shell and rapid-fire broadsides. That was plainly demonstrated in the battle of August 10th. Moreover, there will always be cases where battleships will be called upon to fire at inferior foes, such as cruisers, for instance. As against these medium-caliber guns would be sufficient, and a battleship which no longer carries them would be obliged to allow her large guns to be impaired. That would be against the law of rational economy of forces. The same applies also to actions of battleships among themselves.

The above mentioned technical defects and dangers of large guns will in all probability not be able to impede the military development. On the other hand, the technical arts will endeavor to eliminate and minimize them. Whether it will be possible to render the raw material more durable is doubtful; for it is not the high gas pressure, but rather the enormous heat generated upon the burning of the powder charge, that causes the rapid deterioration of the material. Attempts to reduce the degree of temperature by different chemical compositions of the powder have so far not been successful as far as known. If no decisive progress is made in these two directions there remains but one expedient for prolonging the life of guns, and that is to decrease the quantity of the powder charge. It is true that this would result in a reduction of the

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power of perforation of the projectile, and we shall probably have to put up with that, for the disadvantage of a decrease in perforative power is small as compared with the advantage gained by enabling the gun to withstand a considerably larger number of rounds. New inventions in the field of the technical arts in question may, of course, upset the foregoing chain of reasoning in a minute and put new points of view in its place. In England a definite course has probably already been decided upon. This may be inferred from a recent remark in a service paper, which expresses its satisfaction over the fact that a very high rapidity of fire has been reached, even for the largest guns. It is on the solution of these questions, after all, that the whole new type and its efficiency will depend. It is highly probable that the experience of the *Majestic* class has not failed to produce its effects upon the English naval authorities.

But the more rounds the large guns are able to fire the more difficult and important becomes the question of ammunition supply. We speak now of a coaling strategy—that is, a strategy based on the possibility of keeping the fleet constantly supplied with coal, and the war in East Asia in all its phases has shown the importance of it. Something similar will in future apply also to the ammunition supply, and the replenishing of the ammunition expended from naval bases, or the taking along of ammunition ships will form one of the most important questions of future wars. Another point not to be overlooked is the change in the ship's draught caused by the considerably increased consumption of ammunition. This must, of course, be added to the coal consumption, and the total will be so great that the armor of the waterline will have to be made considerably wider. Even in the present war, Russian battleships, as well as armored cruisers, have sustained leaks below the narrow armor belt as the result of hits, and that will be the case to a much greater extent when a battleship with large, uniform-caliber guns has consumed large quantities of ammunition and coal. The danger is further enhanced by the fact that a 30.5-cm. projectile which strikes the water near a

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ship has a much better chance than a lighter one of penetrating the water and striking the unprotected part of the ship's side. Whether this difficulty should be met by widening the outer armor belt or by particularly careful subdivision into cells and a thin inner armor below the waterline are technical questions which can be decided only by a combination of theoretical computations and practical experiments.

The outer appearance of these battleships of the future will differ materially from that of our present battleships, especially as regards the upper parts. The central casemate for the medium-caliber guns, which has hitherto occupied the whole central part of the ship, will vanish entirely or in part, and its place will be taken by the heavy-armored turrets for the 30.5-cm. guns. In a certain sense this would mean a reduction of the easily vulnerable surfaces. The large, massive superstructure would disappear, making the target perhaps somewhat lower; but the forward and after conning bridges will probably have to be kept at the same level as at present, and they will also have to be connected with each other by the so-called light flying bridge, which passes above the two turrets between them. The suitable arrangement of the inner spaces will present decided difficulties. Considerable space will be lost, on the one hand, by the absence of part of the upper superstructures, which will be necessary in order to afford the large turrets the requisite arc of fire, and on the other hand, also below deck, where the bases of the turret, which must be carried through to the protective deck, will take up considerable space. Thus the quartering of the officers and men will probably prove very difficult.

From all the foregoing it follows that the displacement of such battleships will have to be very large, and further, that, notwithstanding the very moderate confines of the new naval programe, their cost will be very considerable. An industrial paper has computed the additional cost, as compared with our present displacement of 13,200 tons, at 8,000,000 marks per ship. I am not in a position to verify this estimate in detail, but it is probably not very far out of the way. Our present

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battleships cost about 25,000,000 marks each, so that the increase in cost will amount to about one-third of the sum heretofore required, and will correspond to a displacement of 16,000 tons.

THE FRAHM SPEED INDICATOR.

By Friederich Lux.

The Frahm Speed Indicator is based on the employment of the *principle of resonance.* This, as is well known, is the property that elastic bodies possess of being set in vibration, if they are subjected to rhythmic impulses, the frequency of which corresponds with the natural period of vibration of the bodies themselves.

Resonance may be manifested wherever wave motion occurs, as, for instance, with sound, light and electricity. The effects of resonance in accoustics are well known in connection with musical instruments. The fibers of Corti in the ear are an interesting example of elastic bodies set in vibration through resonance. In the case of wireless telegraphy, one of the most modern scientific developments, electrical resonance plays an important part.

The destruction of the Suspension Bridge at Angers, on the 16th April, 1850, by which 226 French soldiers lost their lives, proved a disastrous instance of the effect of resonance.

The investigations undertaken by Mr. Hermann Frahm on the torsional vibrations in propeller shafts have clearly proved that resonance in this connection may have a most momentous effect. Mr. Frahm, however, did not cease his investigations when he had found a means of suppressing the dangers of resonance, but he went further and developed the idea of usefully employing the same principle for the purpose of measuring velocities. His experiments resulted in the discovery of his present form of speed indicator.

The fundamental part of the Frahm Speed Indicator, of which there is a full size illustration in Fig. 1, consists of a spring of best watch-spring steel or other elastic material, the dimensions for ordinary purposes being 40 to 55 mm. long, 3

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mm. wide and 0.25 mm. thick. These sizes permit of a convenient employment of the apparatus, and are sufficient for most practical cases. They may be varied in accordance with any special application.

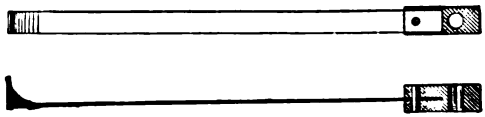

Fig. 1.

The springs are set in a slit cut in a small rectangular shoe, to which they are firmly pinned and soldered. At its upper end each spring is bent over at right angles for about 4 mm., and the head thus formed is covered with white enamel so as to render it easily visible. In the angle which the head makes with the stem of the spring is placed a small drop of solder. The number of vibrations such a spring will make depends primarily on the length of the vibrating part and the weight at the head, as it is made of a material as uniform as possible in quality and thickness. Thus, if the springs have a stem of 40 to 55 mm. long, and more or less solder is attached to the head, any desired period between the limits of 35 vibrations per second and 100 vibrations per second can be obtained.

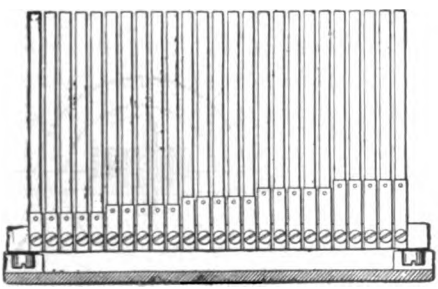

Fig. 2.

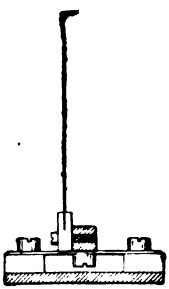

Fig. 3.

An assemblage of such springs attuned to various periods is illustrated in figures 2 and 3. They are screwed to a bridge consisting of an iron or brass rod 6.5 mm. square, and are spaced at a distance of 1 mm. from each other. This arrangement,

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somewhat resembling a comb, forms a self-contained system for certain limits of measurement. The length of the comb is, so to speak, unlimited. It might be made as much as one meter or more, and contain 100 units of attuned springs, but in most cases a comparatively small number is taken, for example, 25 to 50, while in other circumstances 3 or 5 are quite sufficient.

This comb is fixed on two end plates of springy material, screwed to supporting feet, so as to permit the bridge to have a small vibrational movement at right angles to its axis. When only a very small number of springs is used, one such bridge is sufficient, and this, under certain circumstances, may be fixed at one side only. In other cases this elastic support is not necessary, and the comb can be placed directly on the spot where it is desired to measure the vibrational movement.

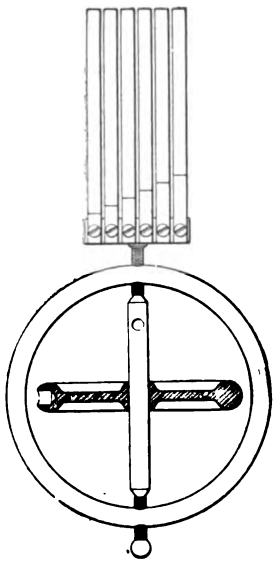

Fig. 4.

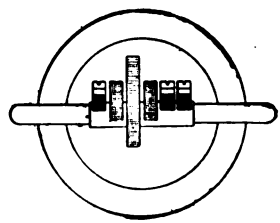

Fig. 5.

The comb can be set in vibration in various ways. The simplest of these is to place it (either with or without the end plates) direct on the frame of the machine, the speed of which has to be measured.

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As it is a matter of difficulty to keep the center of gravity of a rotating system exactly and permanently in the axis of rotation (except, perhaps, in the case of elastic shafts, as in the De Laval turbine), the bed plate and frame of all running machines are subjected to more or less strong vibrations.

These vibrations are in many cases quite sufficient to actuate one of the springs of a comb touching the frame of the machine, and the amplitude of the vibration may reach from 2 to 30 mm., according to the eccentricity of the center of gravity.

A pretty experiment, in illustration of this, may be made with an ordinary gyroscope, by placing a small comb of springs on one of the pivot screws.

With such a cheap apparatus as the gyroscope, it is naturally impossible to guarantee that the center of gravity will lie exactly in the axis of rotation, and, as a rule, it is a good deal out of true, so that the vibration of the springs may have an amplitude of from 40 to 50 mm. Should the center of gravity be very little out of the axis, all that is necessary for the experiment is to cut a small hole in the outer rim of the revolving disc, when a strong effect will be shown on the springs.

Fig. 5 illustrates the appearance of the apparatus as seen from above.

With a machine running at a speed of over 1,000 revolutions per minute, as for instance, a steam turbine, Pelton wheel, fan, or milk separator, a practised eye, accustomed to scientific work in a laboratory, would find an amplitude of 2 mm. or even less quite sufficient for reading off the speed, when the comb is placed directly on it. For practical purposes, where it is necessary to stand two or three yards from the machine, this amplitude would be insufficient, and 20 to 30 mm. at least must be obtained. To do this it is in most cases necessary, except for badly-fixed or irrregularly-running machinery, to have special means for setting the springs in vibration.

The second and the simplest method for the production of vibrations is to employ a tappet wheel or cam.

For example, such a cam, with a number of elevations and

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depressions on its periphery, is fixed on the shaft the speed of which it is desired to measure, and works against a lever, which is thereby caused to vibrate. The vibrations can be transmitted to one of the combs above described by placing it directly on the other end of the lever, or it may be connected with the lever by means of a rod, wire or string.

Fig. 6.

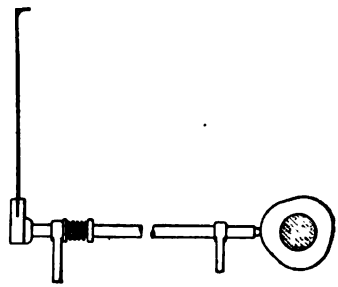

Fig. 7.

In figures 6, 7 and 8 are illustrated these three methods of communicating the impulses to the springs, and by their employment it is possible to make the distance of transmission up to 10 meters.

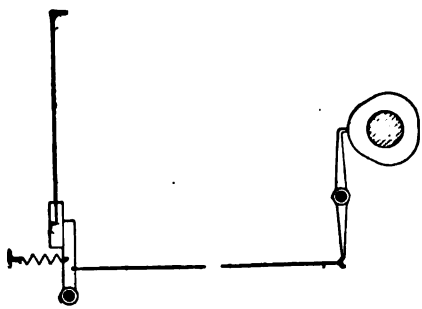

Fig. 8.

Should it be necessary to measure the speed of a machine at greater distances, it is better to employ electrical transmission. The following is one method of affecting this:

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On the bridge of the comb (Figs. 9 and 10) parallel to the springs is fixed a piece of soft iron in the form of a flat bar, which forms the armature of a magnet, on the poles of which two coils are wound.

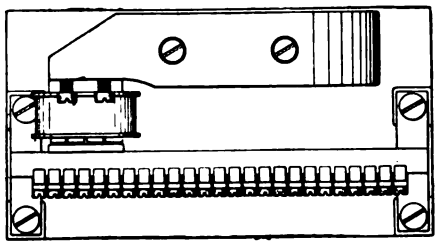

Fig. 9.

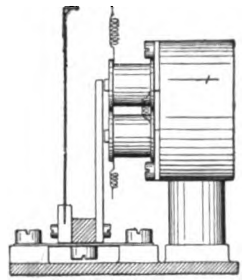

Fig. 10.

If, now, an alternating current is passed through these coils, the armature will be regularly attracted and repelled, so that the comb is made to vibrate in correspondence with the frequency of the current. All the springs are made to quiver slightly at their lower ends, just as above described, and that particular spring whose natural period of vibration is most nearly in unison with the actuating impulses, will be set into very strong vibration, its amplitude depending on the more or less close agreement of its natural period with that of the actuating device.

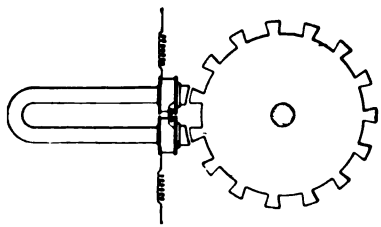

Fig. 11.

If we wish to employ this method to determine the speed of an alternator, all that is necessary is to connect these magnet coils through a resistance to any convenient point in the network. The speed can then be read off directly, as the frequency is a simple multiple of the speed.

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Should it be desired, however, to measure the speed of a machine which does not furnish alternating currents, we must use a special alternating-current generator, which can be driven from the machine in question.

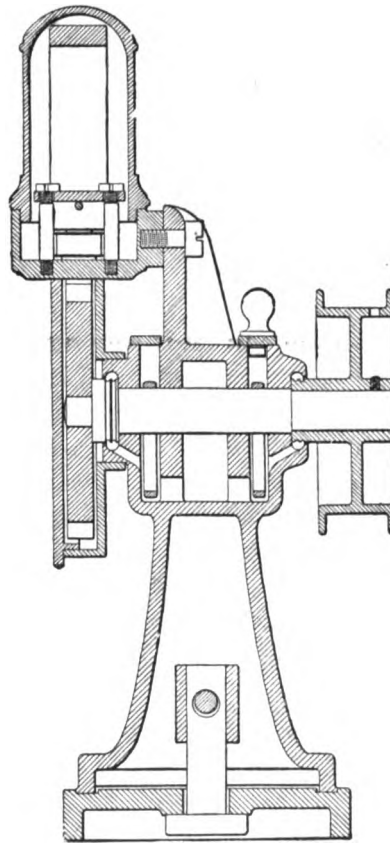

Fig. 12.

A generator of the simplest form for this purpose (Fig. 11) consists of a toothed disc of soft iron, which rotates close to the pole pieces of a permanent magnet, on which are wound two small coils. This revolving armature may either be fixed direct to the shaft, the revolutions of which it is desired to measure, or the driving power may be transmitted from the shaft to the revolving inductor by means of a small belt.

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Figures 12 and 13 illustrate an alternator of the type which we have frequently employed where the revolving inductor can be connected directly to the shaft. This apparatus may be called the transmitter (using the same terminology as in telegraphy or telephony). Its circuit is connected either by

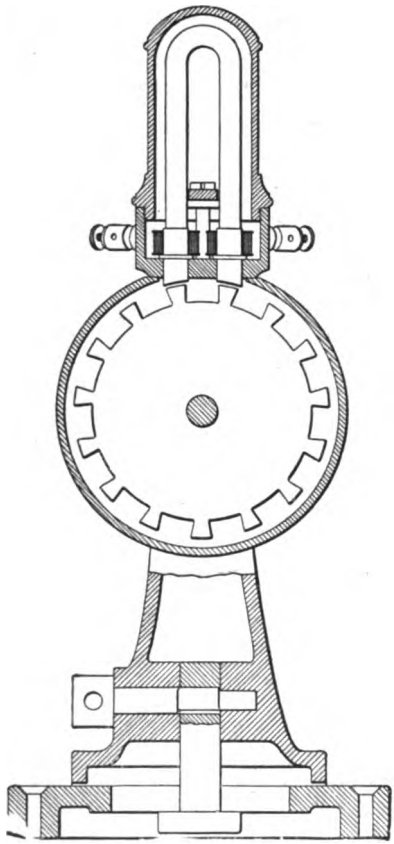

Fig. 13.

two wires or one wire and an earth return to another apparatus, illustrated in Fig. 14, called a receiver, which indicates by its vibrating springs the velocity at any instant.

Instead of an alternating current, an interrupted continuous current can be used to actuate the apparatus, by using one

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of the well-known forms of continuous current interrupters, driven by the machine.

From the foregoing it will be obvious that the vibrator, which is actuated electrically, may be made to serve very conveniently for determining the frequency of an alternating-current or the interruptions per second of a continuous-current circuit, and thereby determine the number of revolutions per minute of any machine or shaft at any distance, either with or without a special transmitter. It will be clear also that one transmitter can simultaneously operate several receivers.

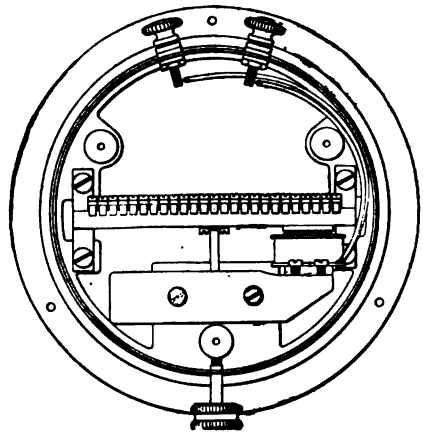

Fig. 14.

Further, as waves of different frequency and amplitude can be superimposed on each other in one conductor and analyzed in the receiver, one receiver can at one and the same time be actuated from several different places. In this way it is possible to compare the frequency of two or more alternating currents.

Frahm's indicator may thus be used for paralleling alternators and polyphase generators, or as a phase indicator, or for the determination of the slip of an induction motor.

It also permits of the observation and supervision at one place of the running of any number of machines, and gives a centralization of the work's management in this respect in a way that was previously undreamt of.

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The long-continued experiments and investigations conducted by Mr. Frahm at the shipyard of Messrs. Blohm & Voss at Hamburg, where he acts as Chief Engineer, have demonstrated that the natural periods of the springs, and therefore the indications of the apparatus, do not vary in the least with the lapse of time, even in the case of apparatus which have been working continuously for nearly two years. Single springs, particularly those which correspond to the normal indication, have during this period made more than one hundred million vibrations, but they have not experienced any detrimental change and possess today their original natural period. The number of vibrations per second is not in the least influenced by the manner in which the impulses are set up, nor by the amount of the movement of the cam, nor by the nature of the electrical transmission. In the latter case the degree of accuracy of the indications is not affected by the variation in the resistance of the conducting wires nor the voltage of the magneto-generator. In fact, the accuracy depends solely on the frequency of the impulses.

Thus Frahm's apparatus embodies the ideal of an absolutely accurate speed indicator, uninfluenced by external conditions and as true in its indications as the natural law which its mode of working exemplifies.

RECENT PERFORMANCE OF THE *TARANTULA*.

The following notes were furnished by Commander R. S. Griffin, U. S. N., member.

On the 9th of May the turbine yacht *Tarantula*, belonging to Mr. W. K. Vanderbilt, Jr., made a short run in Long Island Sound, which was attended by representatives of the Bureau of Steam Engineering of the Navy Department, for the purpose of obtaining information additional to that obtained from the trials of the same yacht last year, and described on page 1095 of Volume XVI. Before starting it was the intention to make several runs at various speeds, but owing to a slight mishap to one of the shaft struts, the extent of which could not

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be readily determined, it was deemed prudent not to make runs at lower than full-power speeds, the mishap referred to having become apparent when about to begin a run at a speed of about 18 knots.

The full-power runs, one in each direction, were made between the Stepping Stones and Execution Rocks, a distance of 3⅝ nautical miles. During both runs there was an ample supply of steam, and, except for the very poor vacuum, a condition noted on her trials last year, the machinery as a whole worked in a very satisfactory manner.

The principal data were:

Speed, knots	23.03
R.P.M. center shaft	1,167
starboard shaft	1,074
port shaft	1,085
Steam in boilers, pounds, by gauge	274
H.P. casing, pounds, by gauge	269
I.P. casing, pounds, by gauge	77.7
L.P. casing, pounds, by gauge	1.8
Vacuum, inches	21.3
Temperature of hot well., degrees	115
Air pressure, in inches of water	4½

Several changes have been made in the machinery since the trials of last year, the most noteworthy being the removal of two Yarrow boilers and the substitution of two of the Mosher type, by which change the grate surface was increased from 66 to 120 square feet and the heating surface from 3,680 to 5,620 square feet. The general design of this boiler follows that of the monitor *Florida*, illustrated in the JOURNAL on page 563 of Volume XV, except that there is a feed-water heater where the baffle of non-conducting material is shown; that the gases pass outward below the line of the heater, and that there are no outside down-comer pipes. One other feature that calls for special mention is the provision for removing tubes. In the upper half of the steam drum there are plugged holes on either side, through which it is possible to remove or replace a tube. The feed-water heater is composed of curved headers, into which nests of tubes are expanded, there being diaphragms in

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the headers to so direct the flow of water that it shall be continuous from the bottom to the top. The water enters the steam drum at the front end and discharges into internal pipes running the length of the drum. These pipes are perforated opposite each tube of the outer row so that the discharge from them will be directed downward and assist the circulation.

One other important change was the substitution of Blake "featherweight" air pumps for the original pumps driven from the center shafts; but this change, as noted above, has not been attended by any improvement in the vacuum. The new pumps are 6 by 12 by 8 inches.

Two Sirocco fans have been fitted instead of the original Sturtevant fan, and an additional auxiliary circulating pump has been connected with the port condenser.

THE CORROSION OF COPPER IN SEA WATER.

We give below a translation of an article written by Herr Uthemann, Privy Councillor to the German Navy, on a number of interesting experiments which this gentleman has been carrying out with a view to discovering an effective means of protecting copper and its alloys against corrosion through the action of sea water.

Copper and its alloys, says Herr Uthemann, have been used in shipbuilding for a long time, and are considered to be valuable materials in the various branches on account of their suitability and adaptability to this industry.

In 1761 the English Admiralty had commenced to protect their ships with copper plates as a preventive to the adhesion of shells to the hulls, and soon afterwards the use of copper for the internal construction of ships increased proportionately to the development of steam engines as driving power. The physical properties of copper and bronze facilitated the manufacture of various parts and fittings both in ships and their engines, and no other material of similar qualities specially suitable for pipe lines was known at that time. Demands for greater speed of vessels were continually made, necessitating

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I.—IN FLOWING WATER.

Number of trial.	Days.	Material.	With or without iron protection.	Increase or decrease per 1 sq. cm. of area in mg.	Remarks.
1 *a*	94	H-copper	With	+ 346.00	Water with 3.5 per cent. common salt. Temperature + 13 deg. C.
1 *b*	94	H-copper	Without	+ 40.25	
2 *a*	52	H-copper	With	+ 411.00	Water with 3.5 per cent. common salt. Temperature + 16 deg. C.
2 *b*	52	H-copper	Without	+ 161.00	
3 *a*	30	E-copper	Without	- 106.50	Water with 3.5 per cent. common salt. Temperature + 16 deg. C.
3 *b*	30	E-copper	Without	— 173.30	
4 *a*	187	E-copper	With	+ 700.70	Water with 3 per cent. common salt and 0.5 per cent. magnesium chloride. Temperature + 18 deg. C.
4 *b*	187	E-copper	With	+ 845.30	
5 *a*	187	E-copper	Without	— 522.70	Water with 3 per cent. common salt and 0.5 per cent. magnesium chloride. Temperature + 18 deg. C.
5 *b*	187	E-copper	Without	— 733.40	

II.—IN STILL WATER.

Number of trial.	Days.	Material.	With or without iron protection.	Increase or decrease per 1 sq. cm. of area in mg.	Remarks.
1 *a*	59	E-copper	With	— 441.30	Bath: Water with 3.5 per cent. common salt and muriatic acid 0.8 per cent. Temperature + 5 — 20 deg. C. Spiral only outside.
1 *b*	59	E-copper-brass	With	— 653.60	
1 *c*	59	E-copper-brass	Without	— 3664.50	
2 *a*	33	E-copper	With	+ 71.80	As under 1.
2 *b*	33	E-copper-brass	With	+ 135.70	
2 *c*	33	E-copper-brass	Without	— 88.30	
3 *a*	198	E-copper	With	+ 267.50	Bath: Sea water 3.5 per cent. Temperature + 18 deg. C. Spiral only outside. The cylinders were also covered before the trial electrolytically with a layer of iron.
3 *b*	198	E-copper	Without	— 553.40	
3 *c*	198	E-copper-brass	With	+ 105.30	
3 *d*	198	E-copper-brass	Without	— 4325.00	
4 *a*	74	H-copper	With	+ 20.35	Bath: Sea water 3.5 per cent. Temperature + 22 deg. C. Spiral only outside.
4 *b*	74	H-copper	Without	— 556.00	
4 *c*	74	H-copper-brass	With	+ 1.19	
4 *d*	74	H-copper-brass	Without	— 422.00	
		Condenser tube from—			
5 *a*	74	Heckmann-bronze	With	+ 39.30	Bath: Sea water 3.5 per cent. Temperature + 22 deg. C. Spiral outside and inside.
5 *b*	74	Brass	With	+ 17.90	
5 *c*	74	Heckmann-bronze	Without	— 497.00	
5 *d*	74	Brass	Without	300.00	

REMARKS.—E-copper = cylinders from electrolytical pure copper; H-copper = cylinders from molten copper, copper-brass = cylinders half copper, half brass, as in Fig. 3.

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reduction of weight. This was obtained by using copper, brass, bronze, &c., for such objects which had hitherto been made of cast iron. Not only were cylinder frames and other castings for boiler fittings and pipe lines after that made of bronze instead of cast iron, but also the framework for pumps, condensers and other machinery.

It is significant that at the same time a more pronounced corrosion of thin-shelled brass condenser tubes was noticed, and the Admiralty officials of several nations endeavored to find out the cause of this. Of course, it had been ascertained more than twenty years ago that such deterioration of copper plates which had been attached to sailing vessels and other wooden ships since 1780 took place, but the percentage of such corrosion had increased five to ten fold after 1883. In the "Metallurgy of Stoelzel," Vol. I, "Production of Metals," it is stated: The deterioration of the metal occurred simultaneously with the increased import into England of foreign ore, which necessitated the use of a larger proportion of lead in the melting process. The mixture thus obtained did not stand the subsequent annealing so well, owing to the quality of the copper being impaired. Davy tried to protect the copper plates from oxidation by bringing them into contact with other metals of a more electro-positive nature, such as zinc, wrought or cast iron. He found whenever the area of the protective metal was as $\frac{1}{40}$ to $\frac{1}{150}$ of the copper plates the latter showed neither corrosion nor loss in weight. But it happened that some carbonate of lime and of magnesium occasionally settled down on the surface which subsequently caused fouling by seaweed and shells, thus reducing the speed of vessels considerably. When the ratio of area of protected to unprotected surface was diminished to $\frac{1}{150}$ to $\frac{1}{1000}$ no calcareous or other sediment was observable, and the copper remained, practically speaking, perfectly clean. Although the above process showed satisfactory results during several trials, it was not generally introduced into shipbuilding, and could not have proved to be practical in other respects. Later on Bequerel repeatedly examined the Davy process in his laboratory, as well as on ships in the port

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of Toulon. According to his—Bequerel's—opinion, the failure of the previous trials was due to the fact that the protective metals had to a great extent disappeared, and not sufficient attention had been paid to the disadvantages resulting from the destruction of the oxidizing metals.

Since 1880 it has principally been the condenser tubes which have been destroyed, and later on the copper sea-water mains and other parts of machinery coming in contact with sea water, as, for instance, the circulating-water pumps had these failures, and the difficulty of finding a prompt remedy reduced the use of copper and its alloys to a certain extent; the mercantile marine even went so far as to employ iron more and more, in spite of this material having to be replaced more frequently than copper. A large ocean liner, for instance, which previously used about 35 tons of copper, now only used 10 tons.

As mentioned above, the cause of the corrosion of copper through sea water was explained by its impurities, and specifications only allowed 0.6 per cent. of foreign matter in copper. At the same time various alloys of copper were tried, all with negative results. But even a greater purity of copper did not improve the position. On the contrary, copper made in the ordinary way, and containing a certain percentage of copper protoxide, showed less deterioration than that made by the electrolytic process. From experience we know that very much depends on the composition of sea water and its temperature. In addition to there being a greater percentage of sodium chloride there is an additional proportion of magnesium chloride which seems to act deleteriously, and it has been proved that tropical temperature as well as artificial heating in condensers, &c., especially favors corrosion. Bronze fittings in pipe lines have the tendency to corrode copper with which they come into contact, or to destroy soldered places. The neighborhood of iron also produces pitting, as can be seen whenever copper pipes come in contact with iron plates of the bilge.

The idea that the chemical decomposition of copper in sea water is furthered by electrolytic action seems to be generally

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confirmed. It is, however, very unlikely that sea-water mains are in any way influenced or damaged by the action of electrolysis caused by leakage from the electric light and power wires.

Trials to counteract the damaging action of sea water have repeatedly been made, the metals being covered with india-rubber, asphalt, marine glue and other elastic lacquers. One of these, invented by an American named Sabine, has been introduced, and proved satisfactory. Another method which is more widely preferred is the equipment of condensers and pipe lines with zinc protectors. All these methods, however, do not give sufficient protection; in fact, they have so many secondary failings that their general introduction has been restricted. The inside covering of rubber, pitch, &c., can hardly be made sufficiently smooth and free from cracks, and the examination of the same presents considerable difficulty. If part of the tube remains uncovered corrosion will act more intensely at that place. These coverings also decrease the bore of the tube, and with changes of temperature small pieces might drop off and choke the passage. Zinc protectors only act within limited distances, and require to be often renewed. It may also happen that small pieces are detached, and damage valves, &c.

Other experiments intended to find a substitute for copper to be used for condenser tubes and bolts, &c., were made, and copper bronze (97 per cent. copper, 3 per cent. tin), Duhre bronze, and a great number of other copper alloys have been tried. The English Admiralty, in the beginning of the eighth decade of last century, tried to stop quick corrosion of bronze condenser tubes by specifying condenser frames and tubes to be made out of the same mixture, the one being cast and the other drawn. The prevalent idea was that the cause of the destruction must be attributed to electrolysis set up by the use of different metals.

The destructive action of sea water on copper is, however, in the first place, a chemical process, as can be proved by exposing alloys of copper with tin or zinc and other metals in

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heated and condensed sea water where they will dissolve. Shortly after immersing the metal in sea water the latter becomes intensely blue, and the metal rod continually loses in weight, and, as can be seen from the annexed table, the loss is often very considerable. Rolled material with a high proportion of zinc does not dissolve so quickly, but in this case also the liquid shows quite distinctly some bright thread-like separations. The manufacture of a copper alloy which would stand casting, rolling and drawing equally well is a very difficult problem, but, even if solved, it would not produce condenser tubes incorrodible in sea water.

The tinning of tubes, as it is now carried out, offers a certain protection, provided the inside tinning is faultless and free from cracks; but this process, again, has the disadvantage of making the tubes very soft, owing to the tin bath forming an alloy with the tube material, no matter whether the latter is copper, bronze or brass. The temperature of the tin bath must be kept low by an addition of lead, otherwise the material will lose weight. The tin covering does not last longer than a year, and when renewed a certain percentage of the tubes is spoilt.

Some years ago sea-water pipes of copper and iron were lined inside with a layer of lead, which is impervious to sea water. Such layers cannot well be made of less thickness than 3 mm., consequently the tubes, on account of this lead layer, are heavy, and can only be bent slightly. Lead-lined tubes, therefore, are less suited to meet the difficulties than cast-steel tubes. The trials of ship plates by Davy, mentioned above, and the general experience that condenser tubes of brass and copper are little attacked in cast-iron condensers, suggested the expediency of making a trial to see if it were possible to counteract the chemical action of sea water by strong electrolytic action. As the latter can only be exercised within limited distances, it was thought advisable to spread the protector, in the form of a spiral, over the copper area, and to use a more solid and more electro-positive metal, such as iron, steel, &c., instead of zinc, which dissolves in water. In immersing a copper tube thus

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prepared in sea water, a strong action producing bubbles and a yellowish coloring of the water sets in immediately. At first the iron is strongly oxidized, whereas the copper is not changed either in color or brightness. After some time the copper surface is covered between the coils of the spiral with a deposit of iron oxide, which soon adheres so firmly that it is able to withstand the action of flowing sea water. This layer of iron oxide, in fact, becomes so attached to the surface of the copper that, after the removal of the iron spiral, it cannot be wiped off with water, spirits or benzine. If it be removed by means of a scraper or some other sharp tool, the clean and untarnished copper surface appears under the brown layer of oxide of iron. Under the iron spiral itself the surface remains for some time almost unaltered, and the corrosion of the iron spiral is diminished as soon as the uncovered surface of the copper is covered with a lining of oxide. The protecting spirals consequently remain inactive for a long time, and even after their complete destruction the coating of oxide lining forms a protection for the copper against the dissolving action of the sea water. As a result of a great number of similar trials with copper either protected or otherwise with iron, in still or in flowing sea water, it has been ascertained that with iron protection the copper gains in weight, whereas without protection it continually lost weight. A trial with a copper tube lined with iron by the galvanic process showed that this covering dissolved in sea water. The surface of the copper became bare and was attacked. It was further proved that the iron protection not only remained intact with pure copper, but also with copper bronze. Brass and other tubes which consisted of segments of copper and brass respectively, which were soldered together, were tried. The condenser tubes when untinned, and when protected with iron, were found to last better than when tinned. The untinned material has the additional advantages that it is harder and resists better the expanding process of the ends into the tube plate. With iron protection it it not necessary to make the tubes from special copper bronze—97 per cent. Cu, 3 per cent. Z—as tubes made of ordi-

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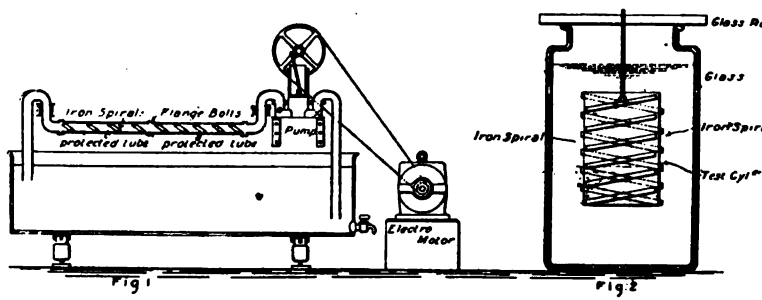
nary alloy, which are not half so expensive, can be equally well protected in the same way. The protection afforded by iron, however, is not confined to sea-water pipes only, but is also applicable to copper pipes fixed in the bilge. These ordinarily are corroded on the outside by the action of the bilge water, which is generally of an acid nature. In this case the iron spiral must, of course, be arranged on the outside. According to a preliminary trial, it seems even to be possible to protect copper coils used in the iron brine vessels of refrigerating plants by covering the former with wire. Trials have



been made in flowing and also in still water. Fig. 1 shows the arrangement used in the former. The tubes A and B undergoing the test were connected with a main through which sea water was pumped by an electrically-driven pump. The apparatus is at work for ten hours daily, and during the remainder of the time the water remains stationary in pipes and tank. The arrangements have been made as similar as possible to those which would be met with on board ship, the pipes being subjected to the joint influence of sea water and oxygen, which are favorable to the destruction of copper. The water is renewed from time to time, as a rule, once in two weeks, and the temperature in the trial room kept to about 20 degrees centigrade. The pumps are made of bronze. In front and behind the trial tubes A and B the main is bent a little upwards, so that there may always be some water in the pipes, which will thus be acted upon when the pumps are at a standstill. The trial tubes were made from copper, manufactured

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both by the electrolytic and the ordinary melting processes. They were properly cleaned and polished inside and outside before use, then flanged, and each carefully weighed. The tests of the protected and non-protected tubes were carried out in similar apparatuses, under exactly similar conditions, and at the same time, consequently the behavior of both pipe lines could be compared directly. Different percentages of salt and other constituents were tried in the sea water, and differences were made in the bolts and flanges. To ascertain whether the protection was only local or was communicated to adjoining parts, the tube A only was protected with iron during one trial. At the conclusion of each trial the wire was removed, and the tube carefully cleaned with water, alcohol and benzine, afterwards being dried and weighed. Trials in still liquid

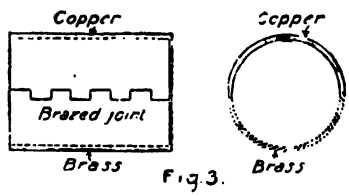

Fig. 3.

were conducted with cylinders made of thin metal in glass vessels, as in Fig. 2. The cylinders were made of copper, produced by the melting process, also of copper bronze, brass and half copper and half brass, as per Fig. 3. These were polished before immersion, carefully cleaned and weighed. Condenser tubes of brass and copper bronze were also subjected to similar trials. The cylinders were tested when protected with iron partly inside only, and also when protected partly inside and outside. The temperature was fixed to 20 degrees centigrade, and the bath consisted of sea water from the Bay of Dantzig, which had been evaporated, so that it contained $3\frac{1}{2}$ per cent. of salt. Subsequently a solution of 3 per cent. of common salt was used, to which $\frac{1}{2}$ per cent. of magnesium chloride had been added. In another case $\frac{1}{10}$ per cent. of hydrochloric acid was added. This was done in order to ascertain whether the protection with iron would also be effective with acid bilge water,

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which, it is notorious, attacks copper tubes vigorously. The trials extended over a period of two years, and the results are shown in the accompanying table. It was proved in all cases that with iron protection, not only copper, but its alloys gained also in weight, whereas without protection copper and its alloys, including those which are called impervious to sea water, suffer a continual decrease in weight.

Considering the great advantages which copper possesses for shipbuilding purposes as compared with all other metals, and that a substitute for it cannot be found, it is highly satisfactory, concluded Herr Uthemann, to have at last found the means to protect the same efficiently. "No doubt," he adds, "the use of copper in shipbuilding, which obviously has been restricted for some years, will soon grow to the right proportions."

It may be mentioned that at the beginning of the action there is a strong formation of rust on the iron protector, and it is advisable to rinse mains for washing and bathing purposes thoroughly before use, so as to get them clear of sea water. The formation of rust, however, only damages the steel wire a little, and in the tests the wire in the tubes had, in spite of its thickness of ½ mm., not lost its elasticity and cohesion after a year's work.—"The Engineer," London.

THE RELATION OF DEPTH OF WATER TO SPEED AND POWER OF SHIPS.

Some interesting data recently presented in "Engineering News" on the subject of change of draught and trim of vessels in motion, showed very clearly that the behavior of a moving ship depends on the depth of water under the keel. Those data suggested also that the power consumption of a vessel at given speed is affected by depth of water. Such a relation is of wider interest than the changes in draught or trim, but hitherto little has been available in the way of definite knowledge about the power relation. An experimental investigation of the subject made a year ago in the German Navy should therefore be interesting, the more so as the experiments were

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made with a full-size vessel (as against model tests), and at such speeds that the reaction of the bottom upon the ship's resistance was felt through a very great depth of water. The investigation is reported in the "Zeitschrift des Vereines Deutscher Ingenieure" of December 10, 1904, by Naval Constructor Paulus, and the following summary is taken from that report.

Two trial trips of a certain torpedo vessel of the German Navy, made on two different measured-mile courses, but otherwise under practically identical conditions, gave speeds differing by about 1½ knots. The slower speed was obtained over the shallower course, having a depth of water of 60 to 80 feet, while the other course was in water nearly 200 feet deep. At first, going on the supposition that a depth of ten times the draught was sufficient to eliminate the influence of the bottom, the authorities were unable to understand this result. However, after a careful re-survey of the two courses, it was decided that no other source of difference existed, and experiments were ordered to investigate the question of influence of depth of water on speed.

The deeper of the two trial courses in question was found to be ideally located for the purpose of the experiments. A flat bottom of very gradual slope enabled all depths from 23 feet to 197 feet to be had in a series of parallel courses, with practical absence of currents. Six courses were laid out, in depths of water respecttively 7, 10, 15, 25, 40 and 60 m. (23, 33, 49, 82, 131 and 197 feet). The torpedo vessel *S119* was used for the experiments. The dimensions of this vessel are not stated. Brassey gives them as: Length, 207.7 feet; breadth, 22 feet; draught, 8.9 feet; displacement, 350 tons.

Six speeds were run, on consecutive days, over the group of six courses. The program for each day was as follows: Beginning with the deepest (197-foot) course, the vessel ran over this course, then turned and steamed back over the 131-foot course, then turned again and went over the 82-foot course, and so on until the 23-foot course had been run. Then it returned over the 23-foot course and worked back in similar

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manner to the point of beginning. This entire double run was made at approximately the same speed, the steam valve being left untouched through the run except where necessary to maintain the speed reasonably uniform.

In a complete run as described, it will be seen that each course was covered twice, in opposite directions; the two results for each course were averaged to eliminate current and wind effects. The change in draught, from the beginning of double run until the vessel returned to the same point, was the only about 1¼ inches, which was almost negligible. Allowance for all differences in draught was made on the assumption that, for equal revolutions, speeds are inversely proportional to displacement, and hourly coal consumption directly proportional to displacement.

The measurements made during the runs were the following: Total revolution of engines; indicated power of all (6) cylinders; mean speed of vessel; trim of vessel, or longitudinal tilt compared with position at rest; shape of water surface along the side of the vessel. The trim was measured by two intercommunicating water gauges separated one-tenth of the vessel's length. The shape of water surface was measured from the deck by graduated rods at intervals of one-tenth the vessel's length.

The results are graphically summarized in the four diagrams, Figs. 1 to 4, herewith. They show (1) the variation of power with depth of water, for a number of different speeds; (2) the variation of power with speed, for six different depths, and in conjunction therewith the change in trim with increasing speed; (3) the variation in speed of revolution with speed of ship; and (4) the variation of slip of propeller with speed of ship. In all diagrams the speed range is the range between 12 to 27 knots, while the total available depth range, 7 m. to 60 m., is well covered by the six curves obtained. The indicated power is seen to vary from 300 H.P. to nearly 6,000 H.P.

The actual speeds used in the six runs were about 12, 15, 18, 21, 24 and 27 knots respectively. Many of the curves of

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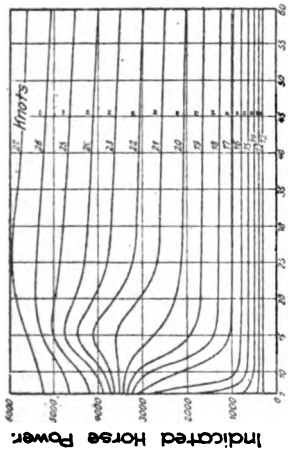

Fig. 1. Variation of Required Power with Depth of Water, for Various Speeds.

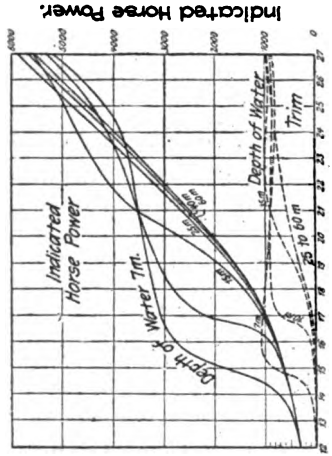

Fig. 2. Variation of Required Power and Trim of Vessel with Speed, for Various Depths of Water.

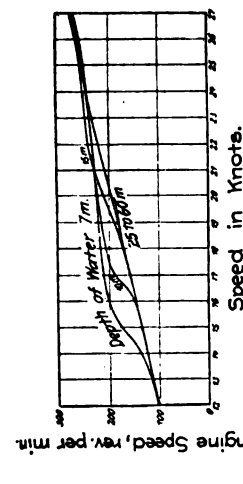

Fig. 3. Variation of Engine-Speed with Speed of Vessel, for Various Depths of Water.

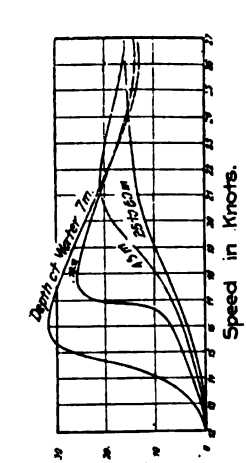

Fig. 4. Variation of Propeller Slip with Speed of Vessel, for Various Depths of Water.

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Fig. 1 are therefore interpolated. In this diagram it will be noticed that curves of three kinds occur. At low speed, as 12 knots, the power is uninfluenced by depth of water (beyond 23 feet). At medium speed the power is very high in shallow water, decreasing rapidly up to 35 or 50 feet in depth, and then remaining fairly constant with increasing depth of water. At very high speed, however, the required power rises as the depth of water is increased from 23 feet, reaching a maximum, and then very gradually decreasing as the depth of water is further increased. The surprising result appears that at high speed the vessel consumed more power in water 200 feet deep than for the same speed in water only 23 feet deep, though at moderate speeds the shallow water consumes several times as much power as deep water.

In the horsepower speed diagram, Fig. 2, there are curves of two kinds, for deep and for shallow water, respectively, of which the former is the type. This deep-water curve rises from a value of about 300 H.P. for a speed of 12 knots, in roughly parabolic form, to a value of about 5,600 H.P. for 27 knots. For any depth, down to about 80 feet (25 m.) the curve is identical in form with this one, and lies only slightly above it. The 15-m. curve, however, shows a radical departure from this type. It begins at practically the same value for low speed, but at about 16 knots it rises suddenly and forms a pronounced hump, whose crest is in the neighborhood of 22 knots. Thence it rises much more gradually, and ultimately it crosses the deep-water curve, so that at very high speeds it calls for less power than the deep-water curve. The curves for depths of 10 m. and 7 m. show the same characteristics, but much more strongly, and at lower speeds in the shallower water. Thus, the 7-m. curve leaves the deep-water curve at a speed as low as 13 knots, reaches a crest at about 16½ knots, and again crosses the deep-water curve at 23 knots, showing lower values than any other curve for speeds beyond 23 knots. The enormous jump in required power as the speed is increased from 13 to 16 knots (500 horsepower to 2,800 horsepower) is especially striking.

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The curves of trim in Fig. 2 correspond generally with the power curves. In deep water the stern sinks gradually and progressively, while in shallow water the stern sinks very suddenly at a point corresponding to the jump in the power curve. Similar characteristics appear in Figs. 3 and 4, showing engine speed and slip of propeller. However, these curves merely supplement, on the theoretical side, the power curves of Figs. 1 and 2.

It should be remarked that the tests here recorded covered six consecutive days, and were made in quiet water and under unusually favorable conditions. The results are, therefore, free from the influence of surface disturbance, as nearly as such tests can be.

The general effect of these experiments is to show that the influence of the bottom upon ships' motion extends through a far greater depth of water than had hitherto been supposed. Further, in such depths as allow of influence of bottom, say, less than 60 feet, there is a certain range of speed which is extremely wasteful of power, better results being obtained with lower and with higher speeds. Experiments upon vessels of other types and larger sizes than the one here used will be necessary to establish the results on a more general basis and to exhibit their dependence on size and shape of vessel. The close relation between required power and trim of vessel, apparent from Fig. 2, suggests clearly that the causes operating to produce change of trim are the same as those which increase the resistance of the vessel.—"Engineering News."

THE DEVELOPMENT OF THE TORPEDO-BOAT DESTROYER.*

In the case of the battleship, as compared with the destroyer, weight is of very much less moment, and it will be seen that the problem of how to give a maximum of strength with a minimum of weight, with a maximum speed with a minimum of coal consumption, must of necessity be the greater task in the destroyer.

* Abstract of a paper read before the Institute of Marine Engineers, by W. J. Harding, on March 13, 1905.

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Torpedoes may be classed as fixed and moving. Of the fixed ones, our experience as a nation was gained during the Russian war of the fifties, when at least one of our ships was slightly injured in the Baltic by the explosion of a torpedo under her.

The moving torpedo, carried on the end of the spar, was used during the Civil War in North America, notably when Commander Cushing blew up the Confederate warship *Albemarle.*

After that we had the Harvey towing torpedo, which was an adaption of the canal barge towed from the bank, followed by the Whitehead invention, which has necessitated alterations and additions both of tactics and of shipbuilding.

The torpedo "drop gear," fitted to vedette boats, etc., was an adaption invented by Engineer Rear Admiral J. T. Corner, and is an arrangement of tongs which is lowered from the boat's side, submerges the torpedo, and, on tripping its regulating valve, thus setting the propeller in motion, releasing the tongs at the same time, the torpedo starts and soon finds its depth. These vedette boats of wood, sometimes called "torpedo boats, wood," in 1878 were 48 feet long and 12 knots speed, have been developed to 56 feet long. Special boats of this class, by J. Samuel White, of Cowes, have obtained 19 knots on trial.

In 1875 Messrs. Yarrow constructed a torpedo launch of the then high speed of 13 knots, fitted with a spar torpedo, and Sir John Thornycroft was the builder of the *Miranda*, in 1872. This vessel was 45 feet 6 inches long, and obtained a speed of 16.2 knots per hour. This firm constructed the first large torpedo boat for H. M. Navy. She was the *Lightning*, completed in 1877, her length 87 feet, speed 18½ knots, and she did many years of constant service at Portsmouth. Many torpedo boats for various governments were successively made by these leading firms.

The precursor of the destroyer class was evolved about 1884. Hitherto the plan had been to give torpedo boats an alternative armament, so that half of the boats might be employed

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for firing torpedoes, the others acting as torpedo-boat destroyers, the boats armed by the small guns resisting attacks of torpedo boats. But as the speeds would be about the same in each, it is evident that neither could decrease the range of the other when attacking or escaping. Perhaps the first approach to the destroyer was the production of Messrs. John Samuel White, of Cowes. She was 150 feet long, fitted with twin screws and with more power—a larger vessel and better sea boat and of higher sea speed than those hitherto in use.

Messrs. Yarrow and Company constructed the *Kotaka* in 1885 for the Japanese Government. She was 170 feet long and fitted with twin screws, and practically an armored destroyer, being fitted with armor 1 inch thick round the machinery and boiler compartments sufficient to resist the fire from the torpedo boats.

About 1893 the "torpedo-boat destroyer" was evolved. Practically these destroyers were given just double the power of the latest torpedo boats in being. The hulls were built round them with due regard to the armament to be carried and the weight to be carried on trial, and also, of course, with due regard to the fact that they would be expected to be more seagoing. The draught was restricted to 5 feet, giving immunity from destruction by Whitehead torpedoes, which are erratic and inclined to "porpoising" if set to run at 5 feet depth or thereabout. They were armed with guns and Whiteheads, and so were calculated not only to destroy torpedo boats, but battleships.

Observers of the building of these hulls will note that a 5-pound plate—that is, ⅛ inch thick—is a very limp object when received from the cogging mills, but after it has passed the leveling slab and been leveled it becomes very stiff; the leveling or "planishing" rendering the plate extremely stiff. The 27-knot destroyers were of various lengths, from 180 to 190 feet, and horsepower varying from 3,700 to 4,800.

One important improvement introduced by Mr. John Samuel White, of Cowes, prior to the year 1883, is a feature in all subsequent torpedo boats and all destroyers. This is the cutting

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away of the "deadwood" at the stern, thereby securing a maneuvering power that would otherwise be scarcely possible, coupled with a more efficient feeding of the propellers so essential to high speed.

In the matter of speed the highest on the measured mile was given by the *Boxer*, by Thornycroft, the speed obtained being 29.1 knots with 4,490 H.P. In the matter of coal consumption the lowest was the *Hornet*, by Yarrow & Co. She burned 9,322 pounds per hour, giving 27.6 knots; the highest burning 17,122 pounds per hour, giving 27.4 knots.

As regards the boilers, it was proved that the locomotive boilers gave excellent results up to a certain grade of coal burning, and at slow burning they were very economical, but the tube ends could not be depended on to remain tight. These troubles were not experienced in water-tube boilers, which were of various systems.

Nevertheless, the locomotive boiler gave 1 H.P. for 1.33 square feet of heating surface, as against a maximum of 2.58 square feet in a water-tube boiler.

I think it may be said that the troubles of water-tube boilers arise primarily from faulty surface condensers, and had satisfactory surface condensers been in vogue when water-tube boilers were first adopted there is no doubt that much public discussion on the great water-tube boiler question would have been avoided.

Trials gave most valuable lessons in screw propellers. It had been customary in torpedo boats and prior boats that propellers should be of forged steel, and the substitution of manganese-bronze gave the then marvellous result of an increase of speed of about two knots an hour, this increase being obtained by using a manganese-bronze propeller of practically the same dimensions as those of forged steel, the figures being that at 384 revolutions per minute a forged-steel propeller gave 24¾ knots. The manganese-bronze one at the same revolutions gave 27 knots in the same vessel at the same draught.

Experiments showed that if only the indicated horsepower

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were required, the propeller could be fitted which would give perhaps 20 per cent. more horsepower than an ordinary propeller. If the speed were required, a very different propeller would be adopted; but if speed coupled with a small coal bill were required, the propeller between these two would give the

DESTROYER, RIVER CLASS, CONTRACT SPEED, 25½ KNOTS.

Destroyer.	By whom built.	Speed obtained on full-speed trials.	Coal per I.H.P. on the high-speed consumption trial	Air pressure on full-speed trial.
		Knots.	*Pounds*	*Inches.*
Welland	Yarrow & Co.	26.2	1.65	1.8
Usk	Yarrow & Co.	26.1	1.77	1.6
Teviot	Yarrow & Co.	25.9	2.07	2
Ribble	Yarrow & Co.	25.8	1.57	1.6
Exe	Palmer's Co.	25.6	2.11	2.4
Waveney	Hawthorn Leslie	25.6	2.19	3.2
Derwent	Hawthorn Leslie	25.7	2.24	2.8
Erne	Palmer's Co.	25.6	2.25	2.5
Dee	Palmer's Co.	25.5	2.28	2.6
Ettrick	Palmer's Co.	25.6	2.33	2.6
Cherwell	Palmer's Co.	25.6	2.34	2.7
Kennet	Thornycroft & Co	26	2.39	3.8
Jed	Thornycroft & Co.	25.7	2.46	4
Itchen	Laird Bros.	25.6	2.46	4.3
Blackwater	Laird Bros.	25.7	2.62	5.3
Arun	Laird Bros.	25.7	2.68	4.4
Foyle	Laird Bros.	25.6	2 79	4.4
Eden (turbine)	Hawthorn Leslie	26.2	*	3.3

*Seven tons and nine hundredweight per hour.

LENGTHS, WEIGHTS AND SPEEDS.

Vessel.	Length.	Speed.	I.H.P. per ton of displacement.	Propellers.
	Feet.	*Knots.*		
Miranda	45½	16	14.7	Forged steel.
Torpedo boat	113	20	10.6	Forged steel.
Torpedo boat	115	21	10.6	Forged steel.
Torpedo boat	130	22	12.9	Forged steel.
Torpedo boat	140	23	15.6	Forged steel.
Torpedo boat	140	23	13	Manganese-bronze.
Torpedo boat	165½	25	15	Manganese-bronze.
River class	220	25½	12.6	Manganese-bronze.
Destroyer	185	27	15.7	Manganese-bronze.
Destroyer	215	30	20	Manganese-bronze.
Destroyer	228	31½	21.1	Manganese-bronze.

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best results. Also that the proportions of the propeller influenced the coal per indicated horsepower, the coal per indicated horsepower propeller lacking in propulsive effect. The least immersion of the propellers gave the best results both in speed and coal bill; and the distance apart of twin-screw propellers was of moment.

I may give comparative instances of propellers showing how improvement was given: (*a*) Diameter increased 4 per cent., area increased 19 per cent., pitch not altered, the coal bill was decreased 27 per cent. at equal high speeds; (*b*) pitch fined 6 per cent., no other alterations, gave the same high speed with 10 per cent. less indicated horsepower and 17¾ per cent. less coal bill; (*c*) diameter increased 2 per cent., pitch fined 9 per cent., gave ¾ knot more speed, and coal bill decreased 40 per cent. at the same high speed obtained by the first propeller.

The terms of the trials for the 30-knot destroyers were very much amplified, for whereas in the former 27-knot destroyers there were no reserves of coal consumption, in these it was intended that 2½ pounds per indicated horsepower per hour should be a standard. This was much less than the average of the former 27-knot destroyers. Messrs. Thornycroft and Company obtained the first record for 30 knots in a British naval vessel at the Maplin Sands.

The 30-knot destroyers were followed by a reaction in favor of stronger vessels, now known as the "River" class. These vessels are of about 550 tons displacement, 200 to 230 feet long, and horsepower, 7,500.

The table from a Parliamentary return gives information of the "River" class. The table will show that in the 20-knot boats of the eighties it required 10½ H.P. per ton of displacement, and so on through each batch of vessels as they became larger.—"The Practical Engineer."

A NEW STEAM TRAP.

The steam trap, of which cut is here given, recently passed a successful test by a board of naval officers at the Norfolk

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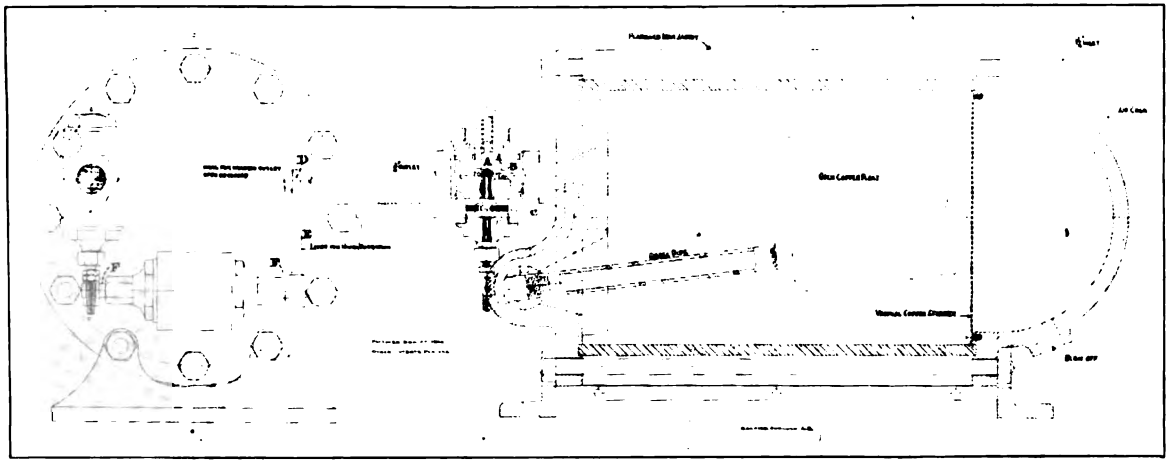

LYTTON STEAM TRAP.

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Navy Yard. It was designed and built by Mr. J. W. Lytton, of Portsmouth, Virginia.

The trap has a cylindrical body of steel or wrought iron and two cast heads, the three parts being bound together with tie bolts, as shown in the cut. One of the heads holds the strainer, the dirt thrown off from which can be blown out through a drain connection. The other head contains the valve mechanism, which is arranged to be easily accessible. When, from accumulation of water in the body of the trap, the bucket becomes filled, it drops, and affords a passage for the water *via* the hollow bucket lever. The same motion trips the auxiliary valve A, which is seated on the main valve B. There results a flow of water down the hollow spindle of the main valve, and this water raises the main valve by acting on the under side of piston C. It is obvious that whenever the trap discharges it must be with a wide opening of the main valve. The auxiliary valve can be held in the open position by locking the arm E on trip shaft F with the pawl D.

The trap can be momentarily by-passed by tripping the auxiliary valve by light pressure on the arm E, the auxiliary valve being practically balanced under all steam pressures.

The trap can be easily and efficiently lagged, is of light weight for a given capacity, has free opening for discharge, may be easily overhauled, is not liable to serious derangement under ordinary conditions of service, may be readily by-passed, can be conveniently attached to a support, and can be manufactured at a low cost.

STEAM SUPERHEATERS—RECENT PROGRESS IN THEIR CONSTRUCTION.

By Maurice Miet.

(Reprint from the "Technical World," June, 1905.)

The principal results developed by the many experiments made during the past forty years in the use of superheated steam have been to bring out the exceedingly delicate character of the superheating apparatus that has been evolved, and to show the difficulty of properly maintaining it, the irregu-

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larity of its action and frequently the ephemeral nature of its life.

So, in spite of the coal saving shown by superheaters, frequently amounting to as much as 20 per cent., the more delicate types have been abandoned. On the other hand, improvements made in recent years, especially by the substitution of cast steel for cast iron and by the elimination of concealed joints, have made it possible greatly to simplify and strengthen the construction. As a consequence of this the use of the superheater has become very extensive, especially in Germany, in connection with engines designed to use steam at temperatures of from 575 degrees to 675 degrees Fahrenheit; and it is thought that the same will soon be true in France, where superheaters had their birth and where they were first used.

GENERAL PRINCIPLES OF CONSTRUCTION.

The general principle followed in superheater construction has been to make them of a nest of tubes to be traversed by the steam, placed either in the fire box of the boiler itself or at some other point in the course of the hot gases; or away from the boiler altogether, where the superheater may be heated by its own fire. In either case the nest of tubes must satisfy the same conditions as far as circulation of the steam is concerned.

It is important to have a surface of sufficient extent to obtain the desired effect without involving the necessity of raising the tubes to too high a temperature regardless of the amount of steam flowing through them. Tubes uniting two collectors of different forms and arrangements ought to be of small diameter, so that the heating surface thus obtained shall occupy a small space, and the steam thus be kept in contact with the walls by which it is to be heated.

In order that the resistance to the passage of the steam may be reduced to a minimum, after having decided upon the diameter of the tubes, it is better that the headers used should be placed in parallel than in series, and that the several headers should be as short as possible. It is quite true that this arrange-

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ment offers the disadvantage of multiplying the number of joints of these tubes upon their two headers; and, to use it, it will be necessary to have either an extra joint or a well-fitted thread.

The tubes of a superheater ought to be placed side by side, with sufficient space between them for the passage of the gases. For superheaters connected in any way with the fire box of a boiler, the volume of whose gases is always sufficient for the work, means should be adopted to regulate the heat of the nest by the use of a damper permitting a variation in the quantity of gases allowed to pass. In superheaters having an independent fire box it may be necessary to dilute the gases coming from the fire by an excess of air—which improves the combustion while lowering the temperature of the products of the same—so as not to expose the tubes of the first row to the dangers of burning. The natural result of this arrangement is that when these gases leave the superheater they still carry with them a considerable quantity of heat that it is well to utilize in an economizer.

In the above cases some sort of apparatus should be provided for the guidance of the fireman in the adjustment of the damper and the regulation of the fire. For this purpose, and for some years past, a thermometer has been placed in the steam at the point where it leaves the superheater. But the reading of this thermometer merely indicates the effects of the work of the fireman, and fails to indicate in advance what he ought to do, so that it acts rather as a controller than a guide. Furthermore, in case of a considerable diminution or an actual stoppage of the flow of steam, this thermometer placed really outside the superheater cannot indicate what is going on within, where the tubes may become red hot without so much as attracting the attention of the fireman. Such an arrangement of the thermometer is, therefore, insufficient, at least where the work can vary to any great extent and through periods of variable duration.

Better results have been obtained by the observation, not of the temperature of the steam, but of the hot gases at some con-

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venient point in their course. In fact, by taking into consideration the fall in temperature of the gases in their passage over the nest of tubes of the superheater it can readily be ascertained whether the amount of heat given up by them is proportioned to the steam delivery—that is to say, whether the fireman is doing his work properly. In this it has been found that the temperature taken at any point whatever in the course of the gases should remain practically constant for different amounts of steam delivery.

Again, observations of practical workings show that, on account of the mass of masonry which surrounds the fire box, and which is an essential feature in the construction of a superheater, the temperature of the gases, as they reach the nest of tubes, changes very slowly in spite of any sudden opening of the doors for firing or other similar influences. At the same time, the temperature of the gases leaving the superheater is very even, or, at least it changes very slowly through rather narrow limits.

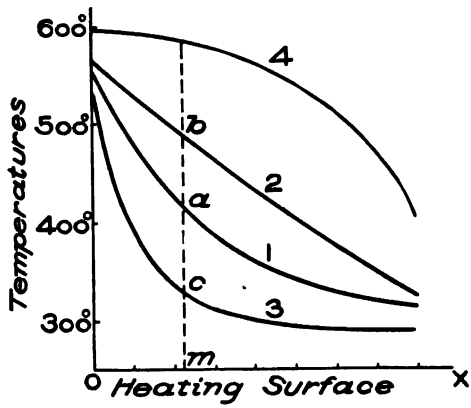

Fig. 1.—Curve Diagram of Temperature of Gases.

This does not hold true, however, in the space between the terminals; and, if we lay out the heating surface of the apparatus on the axis *OX* (Fig. 1), and indicate the corresponding temperature of the gases by the ordinates at each point, a curve like *1* will be obtained if the work of the superheater is good;

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one like *2*, if the heat is excessive; and one like *3*, if it is insufficient. In fact, in the last case the first rows of tubes are quite able to absorb the greater part of the heat furnished them, while the back rows do not work at all. Finally, if the flow of steam is stopped, while the heat continues unchecked, a curve like *4* is obtained, wherein the gases are no longer cooled by the nest, but merely to a slight extent by the masonry.

This figure, obtained experimentally with a large superheater of recent construction, shows that at some point in the surface, such as *m*, there is a region in which the variation of temperature *mb* and *mc* is at a maximum; and that if a thermometer were to be located there in such a way as to maintain the indicated temperature near *ma*, it would forestall any variation of steam output, instead of following it. In fact, the steam temperature only follows the fire variations after some time has elapsed, the metal of the tubes forming a sort of calorific balance wheel, as it were.

LE CHATELIER PYROMETER.

The Le Chatelier pyromerer with a galvanometer attachment and a dial reading directly up to a temperature of 1,100 degrees Fahrenheit is the most convenient thermometer to use

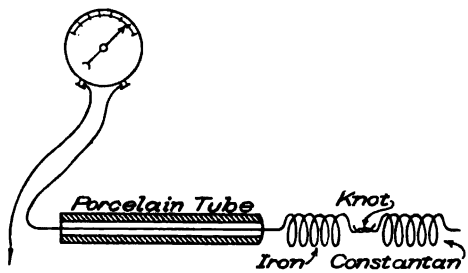

Fig. 2.—DIAGRAM OF LE CHATELIER PYROMETER.

for this purpose. This instrument (Fig. 2) makes it possible to observe the work of the tubes very easily, throughout the several sections of the apparatus, by admitting that the work of transmission of heat is proportional to the excess of tem-

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perature of the gases above the metal at any particular point. This excess, in fact, is measured as follows:

A wire of iron and one of *constantan* (a special metal used in these pyrometers) are united with a twisted knot. The iron wire is passed through an insulating tube of earthenware or porcelain, and fastened to one of the terminals of a galvanometer of the proper sensitiveness, while a second iron wire leads from the other terminal to the piping of the superheater.

The end of the tube is then introduced into the chamber containing the nest of the superheater, and the wire of constantan brought into contact with one of the tubes, when a deflection of the galvanometer needle is obtained. The temperature of the knot is supposed to be the same as that of the medium in which it is placed, and the point of contact with the tube that of the tube itself, when the differential electromotive force which appears will be proportional to these two temperatures.

It would seem, then, that the use of an electric pyrometer, properly placed in the course of the fire-box gases, would serve to protect the tubes against being heated to redness, and thus preserve them from undue deterioration.

MODERN IMPROVEMENTS.

In the old superheaters a system of natural circulation was used, which possessed the disadvantage of exposing the tubes to overheating from the very hot gases when they were not being cooled, on account of the high temperature of these gases as they flowed over them. At present, comparatively fresh steam is used at the entrance, so as to take up the heat of the gases as they leave the fire box, and thus to protect the first rows of tubes.

Other practical details have also been subjected to important improvements, such as the means of removing and replacing the tubes, cleaning them, the construction of special piping for high temperatures, the jacketing, etc. These improvements, made in the construction of superheaters, have resulted in their use in service, where, besides the saving in fuel, they also effect

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a great saving in the condensing and feed waters, besides permitting a reduction in the diameter and weight of piping and valves and avoiding all entrainment of water.

In the matter of the wear and lubrication of engines, oils are now available that will preserve their lubricating qualities at a temperature of 575 degrees Fahrenheit; while the dry lubricants—such as graphite, sulphur and talc—are available for higher temperatures, besides adding no oil to the water of condensation, and consequently none to the boiler feed.

The preceding principles have been applied all along the line by the builders of superheaters, with many interesting variations.

It is not the intention to enter upon an historical sketch of superheating, which would be for the most part a recital of the names of builders whose apparatus is no longer in existence, but to present those whose apparatus is today at work, and to whom the development of the present industry is due.

It may be well, however, to mention the apparatus of Becker (1827); Trewithik (1828); Raffard (1848); De Quillac (1849); Moncheul (1850); and, above all, Hirn (1855), whose work demonstrated beyond all peradventure that superheating does effect an actual and important saving.

UHLER SUPERHEATER.

The Uhler superheater, recently brought out in Germany, is of an old model, with gas tubes in a cast-iron box, which has given way to numerous improvements.

SCHWŒRER SUPERHEATER.

Before speaking of the multitubular superheaters in use, attention should be called to the Schwœrer apparatus shown in Figs. 3 and 4, which has been extensively used with very few failures. This design is directly derived from that of Hirn. It is formed of a series of cast-iron tubes traversed by the current of steam and provided with a double row of inner ribs like those of the Serve tube, but less pronounced; while on the outside there are transverse ribs like those of a radiator. The

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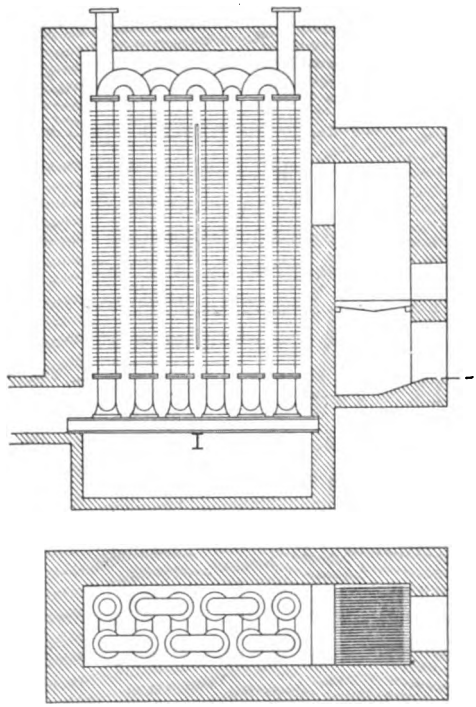
builder has thus secured an amount of heating surface within a given space, which, while not up to that of the most recent superheaters, is, nevertheless, a great advance over that of Hirn. Such an apparatus, when made of hard cast iron, will carry a pressure of 170 pounds per square inch.



Figs. 3 and 4.
VERTICAL SECTION AND PLAN OF SCHWŒRER SUPERHEATER.

GEHRE SUPERHEATER.

The Gehre superheater is formed of one or more cylinders, whose flat ends are perforated with holes, and stayed by a nest of tubes like a locomotive boiler. The fire-box gases pass through these tubes, and the steam circulates around them. The loss of weight which the steam experiences in this apparatus is insignificant, while it does require a decided increase of draft.

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DECK SUPERHEATER.

The Deck superheater resembles the Schwœrer apparatus, except that the external ribs on the tubes are longitudinal, thus increasing the passage for the gases, and slightly decreasing the bulk of the whole.

RECENT GERMAN DESIGNS.

During recent years a great deal of work has been done by German designers, among whom the more prominent are Schmidt, Walther, Gohring, Steinmüller, Petry-Dereux, Durr, Buttner, Rodberg, Schwartzkoff, Hering, Guillaume and Wolf. It will be quite impossible, within the limits of this article, to describe the apparatus of all of them, so that attention will be limited to two—namely, Schmidt and Hering, whose devices have been extensively used in Germany and Austria, as well as to some extent in France.

THE SCHMIDT APPARATUS.

Schmidt was the first who attempted to raise his temperature above 475 degrees Fahrenheit, which had previously been considered the highest limit of practical working. He conceived the idea of raising the temperature of the steam to 675 degrees Fahrenheit, a point at which good oils still remain intact.

The Schmidt superheater (Figs. 5, 6 and 7) is the first of a type having long, parallel, bent tubes, connecting two parallel headers. With this device the superheating improves as the flow of steam increases, for whatever is lost in pressure is gained in temperature. In order to avoid the disadvantages of a fixed circulation and relieve the first rows of tubes exposed to the hot gases, Herr Schmidt divides his nest of tubes into two parts traversed successively by the steam, and thus sets up two distinct nests with four cast-iron headers. The first nest is traversed by the steam in the same direction as the fire-box gases, and so as to take the damp steam, as it comes from the boiler, into the tubes subjected to the direct heat from the fire box. On leaving the first nest the steam passes

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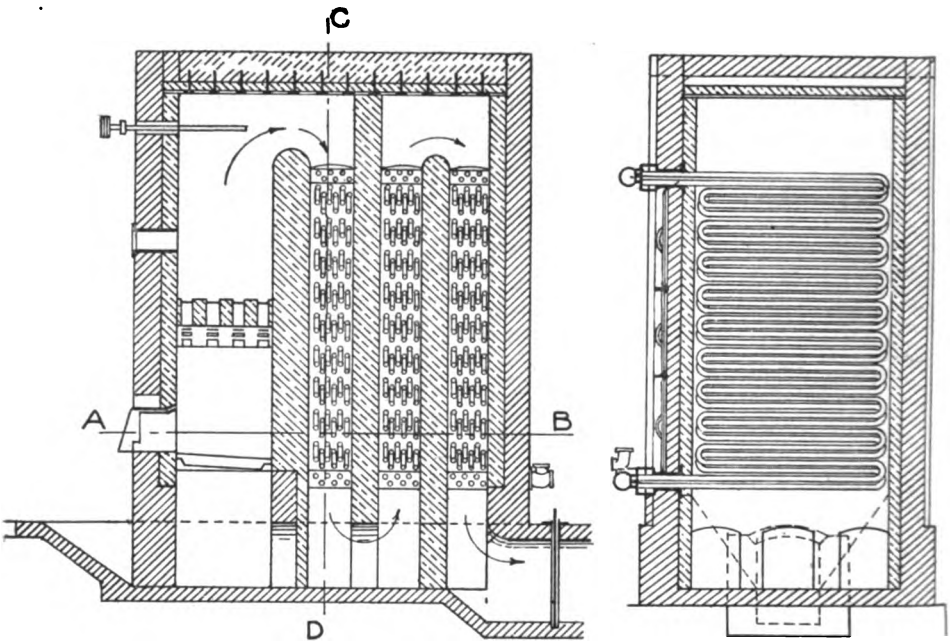
through the second in the regular way. This method has all of the advantages of the regular circulation, throughout the whole of the superheater, if, on leaving the first nest of tubes, the steam has merely been dried and not superheated at all.

Herr Schmidt has designed some arrangements for both boilers and engines intended for the production and use of superheated steam at and above 575 degrees Fahrenheit. He



SCHMIDT SUPERHEATER.

Fig. 5.—Longitudinal Section on E F. **Fig. 6.**—Cross Section on C D.

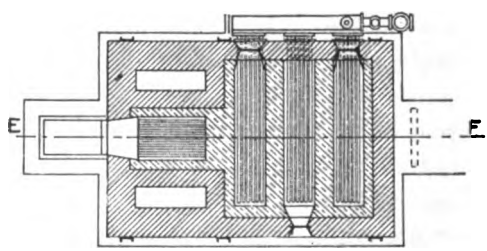


Fig. 7.—Horizontal Section on A B.

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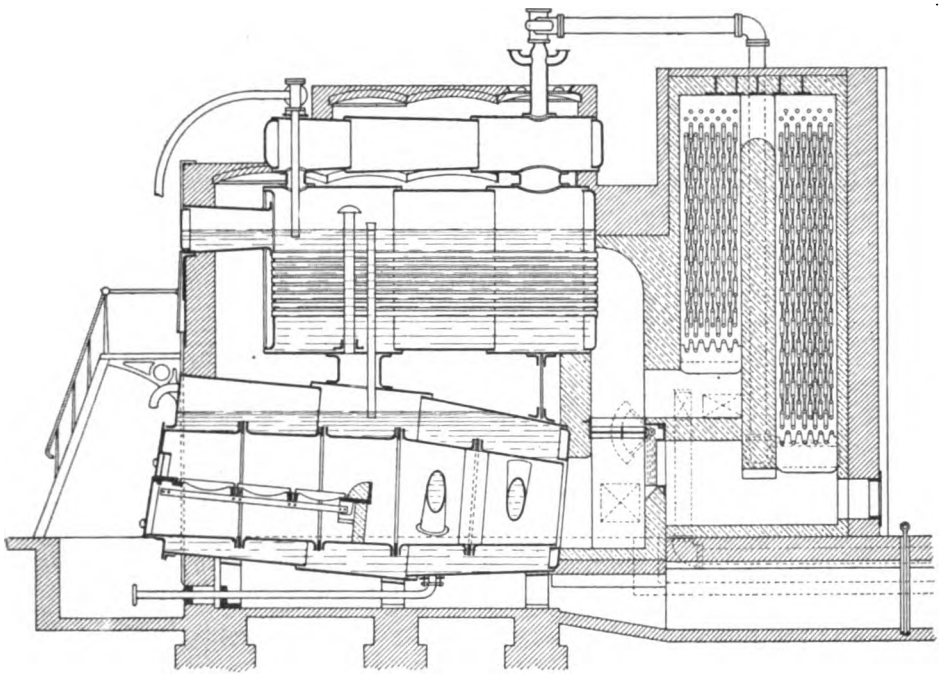

Fig. 8.—Longitudinal Section.

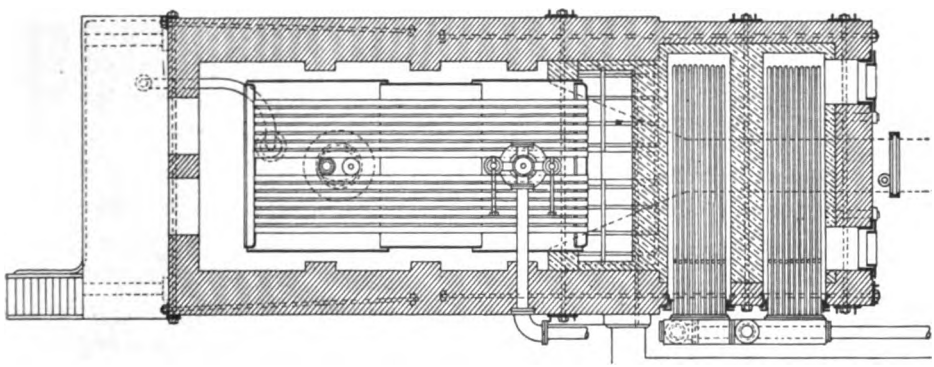

Fig. 9.—Plan.—Boiler fitted with Schmidt Superheater.

was probably the first to realize that in order to produce superheated steam advantageously, it was well to reduce the boiler in the neighborhood of the fire box (Figs. 8 and 9), and thereby

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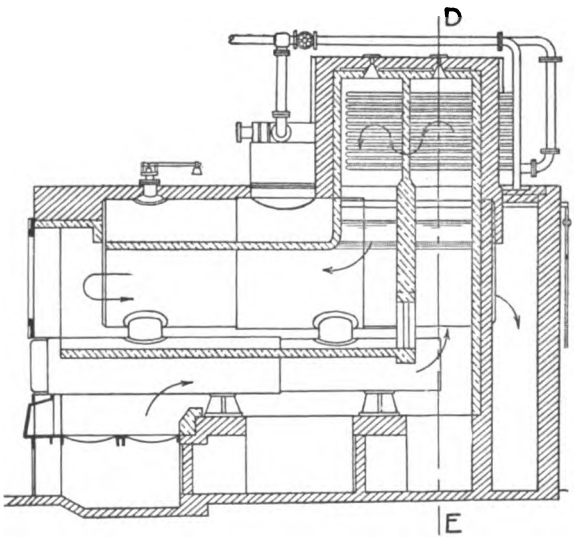
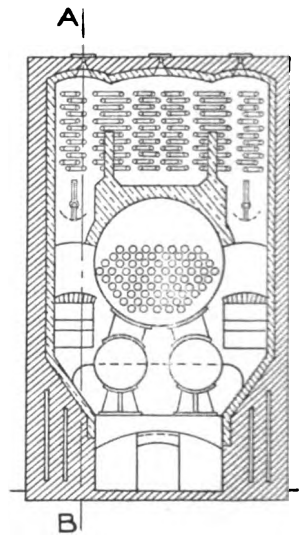
avoid the development of a heating surface each square foot of which becomes less and less efficient according as it is more and more removed from the bridge wall.

It is certainly a rational move to do away with the less efficient portions of the heating surface, and to replace them either with a superheater to which the gases are delivered at temperatures of from 925 to 1,475 degrees Fahrenheit, or with an economizer into which the feed water is pumped, and which, being cold and placed near the outlet, absorbs heat that it would be quite impossible for the boiler to take up.

The Schmidt boilers deliver superheated steam directly, and are built on principles that other makers are beginning to copy. These special arrangements make it possible to apply the Schmidt superheater to locomotive boilers without any essential modification and without interfering in any way with the draft. In the case of stationary boilers of the same class, but having larger tubes, another arrangement permits the placing of the U-shaped tubes forming the nest of the fire tubes themselves without injuring the steam production.





HERING SUPERHEATER.

Fig. 10.—Longitudinal Section on A B. **Fig. 11.**—Cross Section on D E.

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THE HERING APPARATUS.

The Hering superheater (Figs. 10 and 11) is formed of weldless tubes 164 feet long. These tubes are bent into parallel sinuosities, and enter headers at each end in a way to leave the joints perfectly accessible from the outside. An arrangement of dampers makes it possible to regulate the range of temperature within narrow limits. The circulation of the the gases is so controlled that a deposition of soot upon the tubes is avoided, and the number of the latter is such that the velocity of flow of the steam need not be more than 82 feet per second.

MOST RECENT TYPES.

In most recent years the types of practicable superheaters have not been numerous, in spite of the veritable deluge of inventors who have appeared since the Paris Exposition of 1900, and the systems presented have little of interest. Hence only these devices will be cited which have received a practical application, to the exclusion of those subjected merely to laboratory experiment.

First, there is in the old Uhler apparatus, recently revived and remodeled, in which is embodied the original principle of the Field tubes, set up in chambers having a double bottom, but using steel instead of cast iron for the box, and with a conical thread for a fastening. The box is extended out to the masonry by two tubes for the admission and outlet of the steam, with joints that are not accessible (Figs. 12, 13 and 14). The return Field tubes leading to the lower box are no longer subjected to a possible clogging by the scale which may have been entrained, and which would have a tendency to accumulate at the bottom and cause them to burn out. The external cleaning of the tubes is made easy by removal of the false pieces. The circulation of the gases is maintained in such a way that the saturated steam on its entrance serves at the same time to protect the tubes from the direct action of the flames, and to absorb the heat of the gases as they leave, in the regular way. This arrangement lends itself to the use of high pressures and high temperatures. It produces a smaller loss

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of steam pressure than other multitubular superheaters, with the same amount of heating surface.

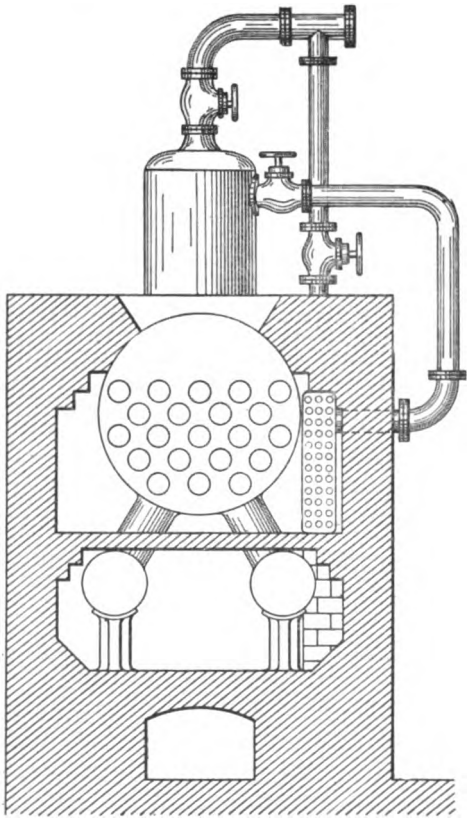

Fig. 12.—UHLER SUPERHEATER WITH HORIZONTAL TUBES.

In the Badere superheater, in order to avoid the complication of the double box or caisson, as in the Uhler apparatus, the Field tubes are not used, but U tubes whose two ends abut respectively into two compartments set side by side and into the bottom of which they are inserted. A common cover of the two forms a joint all around them.

M. Maiche uses tubes of iron or mild steel, as in other multitubular apparatus, but of a larger diameter. The originality of this apparatus consists in filling the interior of the tubes, where the steam passes, with a nest of thin tubes packed closely

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against each other, whose object is to take the place of inner ribs on the Serve tubes forming the heating surface.

At first sight it would seem illogical to provide these tubes with ribs upon the inside only, and not upon the outside; but it should be noted that, under these conditions, the metal approaches more nearly the temperature of the steam than that

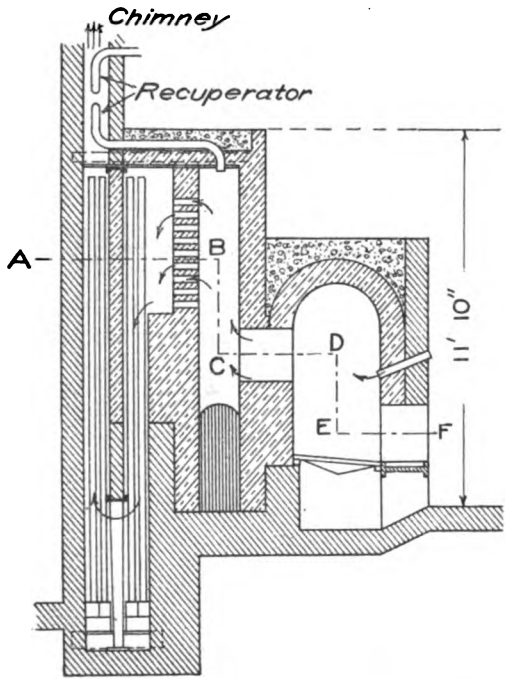

Fig. 13.—Vertical Section on M N.
UHLER SUPERHEATER WITH INDEPENDENT FIRE BOX.

of the fire-box gases. The consequence is that the tubes thus protected are better able to support high-temperature gases, which is an advantage from the standpoint of the preservation of the material and economy of operation by permitting a reduction in the quantity of free air admitted above the grate. The installations of this apparatus in practical operation are not old enough or numerous enough to permit of judgment being passed upon the real value of the results that have been obtained.

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The firm of Piat & Co., of Paris, have recently undertaken the construction of a new German superheater known as the "Schmidt Eindhoven."

A large number of the boilermakers have recently designed arrangements by which nests of superheaters can be attached to boilers. These arrangements, some of which are original, make, as a general thing, no pretense to the use of an independent fire box. The Roser apparatus may be cited as an example of this. It is formed of a thick block of cast iron, surrounding three sides of the fire box above the grate. Within the body of this wall are imbedded the iron pipes through

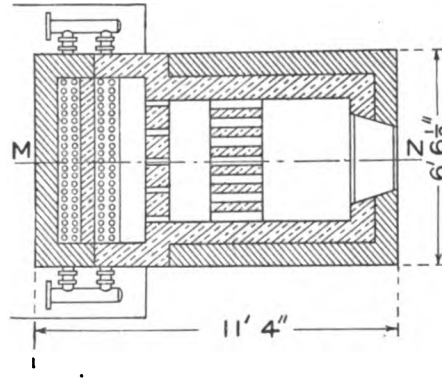

Fig. 14.—Horizontal Section on A B C D E F, Uhler Superheater with Independent Fire Box.

which the steam to be superheated is passed. As this arrangement admits of no regulation of the temperature of the steam, it can merely be admitted that the steam will be superheated to a temperature far above the point needed, so that it is mixed with a definite amount of saturated steam admitted through a special cock. This device is of comparatively recent origin, and has thus far received but a limited application.

This enumeration, although incomplete, shows that the problem of the construction of a practicable and durable superheater has been solved by a number of builders. Without speaking of those new ideas, of which some will undoubtedly be sanctioned by practice, there are several, such as the Uhler,

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Schwœrer, Schmidt and Hering, which have already been put into extensive use, and have shown by a sufficiently prolonged service of ten years or more, that they are durable and reliable and good for many years to come. In Germany, where the most advanced work has been done, there are at the present time more than a thousand plants equipped for the use of superheated steam. And in France the large electric plants of Paris are being gradually fitted with superheaters.

H. M. SCOUT *FORWARD*.

Three of the eight scouts being built for the British Navy have now completed their trials, and in each case the guaranteed speed of 25 knots has been exceeded by quite a quarter of a mile per hour, notwithstanding onerous conditions as to the load to be carried on trial, &c., while the coal-consumption results indicate that the vessels have a greater radius of action than was anticipated. As the only element in their design criticised reasonably had reference to the distance they might travel with their full supply of fuel on board, the general issue is most satisfactory. The first vessel to pass through her trials was the *Sentinel*, built by Messrs. Vickers' Sons & Maxim, Limited. She is now in commission and doing well. The second was the *Pathfinder*, constructed by Messrs. Cammell, Laird & Co., Limited, Birkenhead, and the third, the *Forward*, was constructed by the Fairfield Shipbuilding and Engineering Company, Limited, Govan, Glasgow, and she attained a distinct success in her full-power run on Thursday, May 11th. Each of the firms named has another vessel nearly ready. The two others are being built by Sir W. G. Armstrong, Whitworth & Co., Limited, Elswick, the machinery being by Messrs. Hawthorn, Leslie & Co., Limited, Newcastle-on-Tyne.

For the present we confine ourselves to the *Forward* and her most satisfactory performance. In the course of the full-speed test of eight hours' duration she maintained a mean speed of 25.286 knots on six runs over the measured mile on the Clyde, while the engines were indicating 1,500 horsepower

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less than the designed power. Running at 210 revolutions, 16,500 indicated horsepower was anticipated; on the measured-mile trial the mean revolutions were 211.8, and the power 14,995 indicated horsepower, the links on the twin engines being set to cut off at 58 and 57 per cent. of the stroke respectively. Some explanation for the higher ratio of speed to power than was aimed at in the design is found in the fact that the cruising speed was realized very economically as regards engine power and coal consumption, and that therefore a smaller load of fuel satisfied the condition of the contract, which required that the vessel should, on the full-power trial, carry coal to enable her to steam for 1,500 miles at 10 to 12 knots. The weights, too, evidently came out rather less than was anticipated, and thus the vessel met all the conditions for the full-speed trial on a displacement of 2,790 tons and a mean draught of 13 feet 10½ inches, instead of the estimated displacement of 2,850 tons on a draught of 14 feet 2 inches. But, apart from these facts, the speed and power results indicate a high Admiralty coefficient of fineness and a satisfactory efficiency in machinery and propellers. These latter, with three manganese-bronze blades, made detachable, have a diameter of 12 feet and a pitch of 14 feet 6 inches.

The *Forward* has a length between perpendiculars of 365 feet and a beam of 38 feet 9 inches. She has a considerable rise of floor from keel to bilge, approximating to the spinning-top section, and, in view of recent controversy, has full rather than hollow lines. The wave line was favorable to rapid steaming. She has a moderate freeboard, with a high forecastle and a half poop, of great advantage for sea steaming.

Another feature which distinguishes the *Forward* from some of the other scouts is that instead of having a protective deck she has 2-inch K.N.C. nickel-steel on the broadside from a depth of 2 feet 6 inches under the load line to the top of the side plating, and extending along the machinery and boiler compartments, which is equal to 127 feet of the length of the ship. The total depth of the plating is 11 feet 3 inches. Forward and aft there is a bulkhead of similar thickness, so that the boilers and engines are enclosed within a box with

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2-inch nickel-steel walls. The bulkheads, dividing the boiler space into three compartments, have no openings; there is the usual door in the longitudinal dividing wall between the engine rooms. The ship is lightly armed, having ten 12-pounder and eight 3-pounder quick-firing guns, with two tubes on deck for firing Whitehead torpedoes. Although these vessels may successfully engage torpedo craft, the main function will be not to fight, but to scout in the danger zone and bring information. They lie low in the water, and their pole masts serve only for signaling and for wireless telegraphy.

The machinery, which is in duplicate for twin-screw propulsion, is of special interest, in that it may be said to occupy an intermediate position between those forms which have hitherto been associated with torpedo-boat destroyers and cruisers, respectively; thus the engines, although of the power of cruiser engines, are in general features of "destroyer" design, the cylinders being supported entirely on turned forged-steel columns. The bed plates are of somewhat novel design, and consist of a series of separate cast-steel transverse girders, each transverse girder being provided with one crank-shaft bearing. These cross girders or bearing frames are flat on the under side throughout their whole length, are six in number for each engine, and are connected at their ends by short distance pieces of cast steel. The engines, which are of the type known as four-cylinder, triple-expansion, are balanced on the Yarrow, Schlick and Tweedy system, with the low-pressure cylinders on the outside center line, and the low-pressure moving parts are of considerably lighter scantlings than those of the high and intermediate-pressure cylinders, as is the common practice with engines of this class. The valve gear is of the double-eccentric type, the quadrant links being of the double-bar type, and all joints are made adjustable. The slide valves for the high-pressure and intermediate cylinders are of piston form, and the low-pressure valves are of the flat type with relief rings.

The cylinders, which are all independent castings, and are not fitted with liners, are of the following diameters respectively: High pressure, 31½ inches; intermediate pressure, 50½ inches; and low pressure (of which there are two to each en-

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gine), 57 inches; the stroke in all cases is 2 feet 6 inches. The pistons for the high-pressure and intermediate cylinders are of forged steel and the low-pressure pistons are of cast steel. The covers of all the cylinders are of cast steel. The shafting is hollow, the crank shafts being of 13½ inches external diameter and 6½ inches internal diameter, and the propeller shafts 14½ inches outside diameter, with an internal hole 5 inches in diameter.

The condensers are of rolled brass, built up and riveted together. One condenser is provided for each engine, and the total condensing surface for the ship is 14,022 square feet. Circulating water is supplied to these condensers by 17-inch centrifugal pumps. The air pumps are independently driven, of the twin direct-driven type, a double pump being provided for each set of engines.

Steam is supplied by twelve boilers of modified Thornycroft type, placed in three boiler compartments, and fired from six stokeholds. The aggregate tube surface is 42,960 square feet, and the total grate surface is 752.4 square feet. The working pressure is 260 pounds per square inch.

The conditions of contract required that the *Forward* should maintain for eight hours a speed of 25 knots while carrying in her bunkers sufficient coal to enable her to steam 1,500 nautical miles at from 10 to 12 knots; the amount was to be arrived at by taking the rate of coal consumption on a 96-hours' test. The exact speed to be made on this long trial was left to the discretion of the contractors, who chose the most economical rate. The *Forward* averaged throughout the 96 hours, as shown in Table I, a mean of 10.436 knots, and the mean power was 839 indicated horsepower with the twin engines cutting off at 44 and 43 per cent. of the stroke respectively, and making 80.32 and 80.53 revolutions. On each morning of this 96-hours' trial the vessel returned to the measured mile at Skelmorlie to make six runs; the average speed on these was found to be 10.666 knots for 829 indicated horsepower realized for 82.2 and 82.9 revolutions respectively by the twin engines. The steam results are as follows:

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Table I.—RESULTS OF 96 HOURS' COAL-CONSUMPTION TRIAL OF H. M. S. *FORWARD*, ON MARCH 28, 29, 30 AND 31, 1906, AT 10.436 KNOTS SPEED.

——	Starboard.	Port.
Mean steam in boilers, pounds	142	142
at engines (high-pressure receiver), pounds	49.8	44.8
Mean air pressure in stokeholds	nil	nil
cut off in high-pressure cylinder, per cent	44	43
vacuum, inches	24	24.8
revolutions per minute	80.32	80.53
pressure in cylinders, high, pounds	17.1	15.2
intermediate, pounds	5.7	4.68
low, forward, pounds	1.87	2.64
aft, pounds	1.76	2.69
Indicated horsepower, high	162	146
intermediate	139	113
low, forward	58	82
aft	55	84
Total indicated horsepower	414	425
Gross total indicated horsepower	839	
Coal per indicated horsepower per hour, pounds	2.661	
Water loss per 1,000 indicated horsepower per 24 hours, tons	4.13	
Mean indicated horsepower on miles	829	
Speed, mean of six runs on course, knots	10.666	
96 hours, knots	10.436	

The coal consumption worked out at 2.661 pounds per indicated horsepower per hour, or, as near as may be, to 1 ton per hour. At this cruising speed, therefore, the vessel steams 10½ miles per ton of coal. The vessels have capacity for carrying 500 tons of coal, so that the high economy realized will enable 5,000 miles to be run at 10½ miles, which is much greater than was anticipated when the order for the ships was placed. As the *Forward* has realized her full speed for a low power, her radius of action at 25 knots will be correspondingly increased, and thus she will be the more serviceable in war times. On the full-speed trial the supply to enable her to go 1,500 miles at 10½ knots speed was exactly 145¼ tons. This was the load carried when she commenced her eight-hours' trial on the 11th instant. The displacement was then 2,790 tons, the draught forward being 13 feet, and aft 14 feet 9 inches, giving a mean of 13 feet 10½ inches. The vessel was required to make six runs over the measured mile within one hour and a half of the commencement of the eight-hours' trial, and the results are as follows:

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SIX RUNS ON THE MEASURED MILE AT FULL SPEED.

Run.	Revolutions.	Speed, knots.	Indicated horse-power.
1	213.5	25.568	15,121
2	213.5	25.245	15,301
3	212.5	25.532	14,970
4	211	25.07	14,806
5	210	25.21	14,498
6	213	25.281	15,277
Mean	211.8	25.286	14,995

These results were attained with the engines cutting off at 58 per cent. and 57 per cent. of the stroke respectively. The boilers were steaming easily, the air pressure in the stokehold being 1½ inches, as shown in the first column of Table II, which sets out the mean results from the machinery during these six runs. The second column of the table gives the results for the subsequent 6½ hours of the test.

Table II.—RESULTS OF FULL-POWER EIGHT-HOURS' TRIAL OF H. M. S. *FORWARD*, MAY 11, 1905.

—	Mean of six runs on measured mile.		Mean of subsequent 6½ hours of trial.	
	Starboard.	*Port.*	*Starboard.*	*Port.*
Mean steam in boilers, pounds	250		250	
Mean steam at engines (high-pressure receiver), pounds	227	234	227	234
Mean air-pressure in stokeholds, inches	1.5	1.5	1.8	1.8
Mean cut-off in high-pressure cylinder, per cent	58	57	59	59
Mean vacuum, inches	25.3	24.8	25.2	24.8
Mean revolutions per minute	211.6	212	211.7	209.8
Mean pressure in cylinders:				
High, pounds	109	111.3	112.1	112.3
Intermediate, pounds	34.5	31.9	34	35.3
Low, forward, pounds	14.83	16	15.1	15.8
aft, pounds	14.85	15.3	15	15.54
Indicated horsepower, high	2,727	2,792	2,803	2,782
intermediate	2,224	2,248	2,186	2,247
low, forward	1,217	1,313	1,235	1,282
aft	1,218	1,256	1,226	1,261
Total indicated horsepower	7,386	7,609	7,450	7,572
Gross total indicated horsepower	14,995		15,022	
Total water loss for all 8 hours, tons	13.6			
Speed, knots	25.286			
Loss of water per 1,000 I.H.P. per 24 hours, tons	2.71			

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It will be noted that the engines in the subsequent 6½ hours of the trial averaged over 210 revolutions, so that the speed was well over 25 knots. The machinery worked satisfactorily, while the engine room was more comfortable than is usual with high-speed machinery. From first to last the performance was most favorable. After the trial various other tests as to the maneuvering capabilities of the ship were carried out, and the vessel returned on the 12th instant to Fairfield, to be completed for commission, which will take about two months. The sister ship—the *Foresight*—will leave the same works shortly for speed trials.—London "Engineering," May 19, 1905.

STEAM TESTS OF COAL.

Extract from the preliminary report on the operations of the coal-testing plant of the United States Geological Survey at the Louisiana Purchase Exposition, St. Louis, Mo.

METHODS OF CONDUCTING TESTS.

The method of testing fuels under boilers has been a subject of discussion for many years by the members of the American Society of Mechanical Engineers, and as the result of this discussion a standard method of conducting tests and of reporting the results has received the approval of this society. The steam tests of coal at the Geological Survey testing plant have been conducted and the results reported in accordance with these methods and forms.

The number of tests made was seventy-eight. The duration of each was planned for ten hours, and was as near this time as the conditions at the close of a test would permit. An experienced and careful fireman hand-fired all of the coals tested.

DESCRIPTION OF THE BOILERS.

There were two 210-horsepower Heine safety boilers provided for these tests. They were exactly similar in construction and setting. Each was provided with its own stack, 115 feet high and 37 inches in diameter. Each boiler was fed by its own

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independent injector, and no other means of supplying water to the boilers was provided. The ends of the blow-off pipes were visible during all tests.

In the next table the leading proportions of one of the boilers are given.

The boilers were cleaned externally after each test. The interior condition of each boiler was practically clean during all of the tests.

LEADING PROPORTIONS OF HEINE WATER-TUBE BOILERS USED IN COAL-TESTING PLANT.

Rated capacity of boiler, horsepower	210
Water-heating surface, square feet	2,031
Superheating surface	None.
Grate area, square feet	40.6
Kind of draft	Natural.
Height of steel stack, feet	115
Area of steel stack, square feet	7.67
Number of tubes	116
Outside diameter of tubes, inches	3.5
Steam pressure, pounds	85
Observations taken every 20 minutes.	
Approximate duration of each trial, hours	10

DESCRIPTION OF THE STEAM ENGINE AND GENERATOR.

The steam generated by the boilers was largely used by an Allis-Corliss engine of 250-horsepower capacity, and the power of the engine was absorbed by a 200-kilowatt direct-current, 240-volt Bullock generator, to which the engine was belted.

The engine was the simple noncondensing type. The cylinder was 22 inches in diameter and the stroke was 42 inches. The engine ran about 80 revolutions per minute.

Numerous tests made to determine the steam consumption of the engine, as well as the mechanical efficiency of the engine and generator together, gave the following average results:

AVERAGE RESULTS OF ENGINE AND GENERATOR TESTS.

Number of pounds of steam used per hour per indicated horsepower developed by the engine	23.60
Mechanical efficiency of the engine and generator combined (per cent.)	81

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From these figures has been calculated the electrical horse-power delivered to the switchboard for such tests of the boilers as were not accompanied by a test of the engine and generator.

COALS TESTED.

The coals tested are arranged alphabetically according to the States from which they were obtained. Some of these coals were washed and some were briquetted. In all tests of coals other than slack, the coal was first passed through rolls having an opening of 1½ inches, which crushed it to uniform size. On pages 515 to 517 is a table which gives some of the results of the tests, and in which the character of the coal used in each boiler trial is clearly indicated.

LIST OF COALS TESTED.

Alabama No. 1. First test: Lump and nut coal from mine No. 8 of the Ivy Coal and Iron Company, Horse Creek, Ala.

Second test: Large briquettes containing 7 per cent. of pitch binder.

Alabama No. 2. Lump, nut and pea coal from mine No. 5 of the Galloway Coal Company, Carbon Hill, Ala.

Arkansas No. 1. First test: Lump and nut coal from mine No. 3 of the Central Coal and Coke Company, Huntington, Ark.

Second test: Large briquettes containing 9¼ per cent. of pitch binder.

Arkansas No. 2. First test: Lump coal from mine No. 12 of the Central Coal and Coke Company, Bonanza, Ark.

Second test: Large briquettes containing 11 per cent. of pitch binder.

Arkansas No. 3. First test: Lump and slack coal from mine No. 18 of the Western Coal and Mining Company, Jenny Lind, Ark.

Second test: Large briquettes containing 8.7 per cent. of pitch binder.

Arkansas No. 4. First test: Large briquettes made from slack coal from several Arkansas mines, furnished by the Western Coal and Mining Company, St. Louis, Mo. Briquettes contained 6 per cent. of pitch binder.

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Second test: Small briquettes made with patent binder.

Arkansas No. 5. Lump and slack coal from mine No. 4 of the Western Coal and Mining Company, Coal Hill, Ark.

Colorado No. 1. Run of mine black lignite, from Simpson mine, of the Northern Coal and Coke Company, Lafayette, Colo.

Illinois No. 1. Lump and nut coal from mine No. 1 of the Western Anthracite Coal and Coke Company, near O'Fallon, Ill.

Illinois No. 2. Washed slack coal, from same as Illinois No. 1.

Illinois No. 3. Run-of-mine coal from mine No. 3 of the Southern Illinois Coal Mining and Washing Company, near Marion, Ill.

Illinois No. 4. First test: Lump coal from mine No. 3 of the Donk Brothers Coal and Coke Company, Troy, Ill.

Second test: Same as above.

Illinois No. 6. Run-of-mine coal from shaft No. 1 of Clover Leaf Coal Company, Coffeen, Ill.

Indiana No. 1. Washed run-of-mine coal from Mildred mine of the J. Woolley Coal Company, Mildred, Ind.

Second test: Large briquettes made from washed coal, containing 7 per cent. of pitch binder.

Indiana No. 2. Run-of-mine coal from Electric mine of the T. D. Scales Coal Company, Boonville, Ind.

Indian Territory No. 1. Lump and slack coal from mine No. 1 of the Whitehead Coal and Mining Company, Henryetta, Ind. T.

Indian Territory No. 2. Run-of-mine coal from mine No. 8 of the Rock Island Coal Company, Hartshorne, Ind. T.

Indian Territory No. 3. Run-of-mine coal from mine No. 1 of D. Edwards & Son, Edwards, Ind. T.

Indian Territory No. 4. Lump coal from mine No. 5 of the Western Coal and Mining Company, Lehigh, Ind. T.

Iowa No. 1. Lump and fine coal from mine No. 2 of the Anchor Coal Company, Laddsdale, Iowa.

Iowa No. 2. Run-of-mine coal from mine No. 6 of the Mammoth Vein Coal Company, near Hamilton, Iowa.

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Iowa No. 3. Lump coal from mine No. 4 of the Gibson Coal Mining Company, Altoona, Iowa.

Iowa No. 4. First test: Lump coal from mine No. 3 of the Centerville Block Coal Company, Centerville, Iowa.

Second test: Large briquettes containing 8 per cent. of pitch.

Iowa No. 5. Run-of-mine coal from Inland mine No. 1 of the Inland Fuel Company, Chariton, Iowa.

Kansas No. 1. Run-of-mine coal from mine No. 10 of the Western Coal and Mining Company, Fleming, Kans.

Kansas No. 2. First test: Lump and nut coal from mine No. 11 of the Western Coal and Mining Company, Yale, Kans.

Second test: Same as above, except washed.

Kansas No. 3. First test: Run-of-mine coal from mine No. 9 of the Southern Coal and Mercantile Company, Scammon, Kans.

Second test: Same as above.

Kansas No. 4. Lump coal from mine of the Atchison Coal Mining Company, near Atchison, Kans.

Kansas No. 5. Lump and nut coal from mine No. 11 of the Southwestern Development Company, West Mineral, Kans.

Kentucky No. 1. Run-of-mine coal from Straight Creek mine No. 2 of the National Coal and Iron Company, Straight Creek, Ky.

Kentucky No. 2. First test: Lump, nut, pea and slack coal from mine No. 11 of the St. Bernard Mining Company, Earlington, Ky.

Second test: Large briquettes containing 8 per cent. of pitch.

Kentucky No. 3. Run-of-mine coal from Barnsley mine of the St. Bernard Mining Company, near Earlington, Ky.

Kentucky No. 4. Run-of-mine coal from mine of the Wheatcroft Coal and Mining Company, Wheatcroft, Ky.

Missouri No. 1. First test: Run-of-mine coal from New Home mine No. 1 of the New Home Coal Company, located at Sprague, Mo.

Second test: Large briquettes containing 11½ per cent. of pitch.

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Third test: Same as test No. 1.

Missouri No. 2. First test: Run-of-mine coal from mine No. 8 of the Northwestern Coal and Mining Company, Bevier, Mo.

Second test: Same as above.

Missouri No. 3. First test: Slack coal from mine of the Mendota Coal and Mining Company, Mendota, Mo.

Second test: Same as above, except washed.

Missouri No. 4. Run-of-mine coal from mine of Morgan County Coal Company, near Barnett, Mo.

New Mexico No. 1. Lump and slack coal from Weaver mine of the American Fuel Company, three miles north of Gallup, N. Mex.

New Mexico No. 2. First test: Slack coal from Otero mine of the Caledonian Coal Company, two miles east of Gallup, N. Mex.

Second test: Small briquettes containing patent binder.

North Dakota No. 1. Run-of-mine brown lignite from Lehigh, N. Dak.

Pennsylvania No. 1. Coal from Eureka mine No. 31 of Berwind-White Coal Mining Company, Windber, Pa.

Pennsylvania No. 2. Coal from Eureka mine No. 31 of Berwind-White Coal Mining Company, Windber, Pa.

Pennsylvania No. 3. Small briquettes made with patent binder from anthracite culm furnished by Pennsylvania Coal Company, Scranton, Pa.

West Virginia No. 1. Run-of-mine coal from mine of the Virginia and Pittsburg Coal Company, Kingmont, W. Va.

West Virginia No. 2. Run-of-mine coal from Pitcairn mine of the Pitcairn Coal Company, Clarksburg, W. Va.

West Virginia No. 3. Run-of-mine coal from mine of West Virginia Coal Company, Richard, W. Va.

West Virginia No. 4. Run-of-mine coal from mine of the West Virginia Coal Company, Bretz, W. Va.

West Virginia No. 5. Lump and nut coal from mine of the Davis Colliery Company, Coalton, W. Va.

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RESULTS OF COAL TESTS UNDER BOILERS.

Test No.	Name of sample.	Coal, size and condition.	Chemical composition. Fixed carbon.	Volatile matter.	Moisture.	Ash.	Sulphur (separately determined).	Duration of trial.	Total coal consumed.	Horsepower developed by boiler.	Dry coal burned per square foot of grate surface per hour.	Equivalent evaporation from and at 212 degrees Fahr. per pound of dry coal.	Dry coal per indicated horsepower hour.	Dry coal per electrical horsepower hour.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
			Per ct.	*Per ct.*	*Per ct.*	*Per ct.*	*Per ct.*	*Hours.*	*Lbs.*		*Lbs.*	*Lbs.*	*Lbs*	*Lbs.*
17	Alabama No. 1	Nut, bright	52.52	31.00	2.56	13.92	0.78	10.03	8,656	205.8	20.72	8.44	3.35	4 14
21	do	Large briquettes	50.96	33.00	2.63	13.41	.94	8 25	6,521	196.6	18.97	8.81	3.21	3.96
16	Alabama No. 2	Mine run, bright	48.65	32.98	4.83	13.54	1.17	10.02	9,198	216.4	21.54	8.55	3.31	4.08
9	Arkansas No. 1	Lump, bright	66.36	18.61	1.99	13.04	1.21	10 07	7,071	180.5	16 90	9 05	3.12	3.86
14	do	Large briquettes	67.65	21.21	.94	10.20	1.73	10.03	7,700	206.4	18.74	9 37	3.02	3 73
8	Arkansas No. 2.	Lump, bright	73.65	16.86	1.07	8.42	1.95	10.07	6,517	180.6	15.70	9.73	2.91	3.59
11	do	Large briquettes	60.30	22 49	4.88	12.33	1.32	9.98	7,370	190.3	17.31	9 31	3 00	3.71
29	Arkansas No. 3	Mine run, bright	72.74	16.04	1.97	9.25	1 29	10.02	8,158	219.7	19.68	9 50	2.98	3.68
35	do	Large briquettes	62.04	17.35	2.00	18.01	1.41	10.08	8,600	200 8	20.49	8.34	3.39	4.19
40	Arkansas No. 4	do	71.98	14.06	3.85	10.11	1.64	9.95	8,142	214.5	19 41	9 40	3.01	3.72
42	do	Small briquettes	66.03	18.60	3.18	12.19	1.70	7.03	5,238	166.8	17.78	7.98	3.54	4 38
41	Arkansas No. 5	Nut, bright	73.68	12.54	2 22	11.56	2.39	10 13	9,000	202.4	21.42	8 04	3.52	4.34
75	Colorado No. 1, lignite	do	39.00	35.85	19.78	5.37	.42	9 97	8,972	151.0	17.80	7.21	3.92	4.85
18	Illinois No. 1	Nut, dull	38.21	36 91	9.69	15.19	4.40	9.93	11,124	211.4	24.90	7.21	3.92	4.85
19	Illinois No. 2	Slack, dull, washed	41.72	37.77	10.45	10.06	3.36	9.97	10,090	210.2	22.36	8.00	3.54	4.38
38	Illinois No. 3	Nut, dull	48.75	31.19	8 51	11.55	1.50	10.13	9,537	200.6	21.23	8.04	3.52	4.34
48	Illinois No. 4	Lump, dull	41.59	33.48	13.47	11.46	1.28	10 07	9,367	172.0	19.84	7.37	3.83	4.73
50	do	Nut, dull	43 63	32.44	12 58	11.35	1.36	10.02	10,750	197.8	23.13	7 27	3.89	4.80
73	Illinois No. 6	do	39 62	32.31	13.19	14.88	3 48	9.92	10,343	194.3	22.34	7.40	3.82	4.72
71	Indiana No. 1	Large briquettes	43.23	38.79	11.74	6.24	2.03	6.60	6,579	222.6	21.70	8.73	3.24	4.00
68	do	Nut, dull, washed	40.41	35.17	16.59	7 83	2.18	9.93	10,819	222.7	22.39	8.45	3.34	4.13
65	Indiana No. 2	Nut, dull	40.40	38 04	9.11	12.45	4.12	10.13	9,274	193 4	20.51	8.02	3.53	4.35
10	Indian Territory No. 1	Mine run, bright	46 30	33 96	7.65	12.09	1.80	9.75	8,210	194.7	19.17	8.64	3.27	4.04
20	Indian Territory No. 2	Nut, bright	50.31	36.21	3.71	9.77	1.39	10.17	9,213	212.9	21.50	8.42	3.36	4.15
32	Indian Territory No. 3	do	47 58	37 30	4.79	10.33	3.93	10.02	9,145	205.9	21.43	8.17	3.46	4.27

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RESULTS OF COAL TESTS UNDER BOILERS.—Continued.

Test No.	Name of sample.	Coal, size and condition.	Chemical composition. Fixed carbon.	Volatile matter.	Moisture.	Ash.	Sulphur (separately determined).	Duration of trial.	Total coal consumed.	Horsepower developed by boiler.	Dry coal burned per square foot of grate surface per hour.	Equivalent evaporation from and at 212 degrees Fahr. per pound of dry coal.	Dry coal per indicated horsepower hour.	Dry coal per electrical horsepower hour.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
			Per ct.	*Per ct.*	*Per ct.*	*Per ct.*	*Per ct.*	*Hours.*	*Lbs.*		*Lbs*	*Lbs.*	*Lbs.*	*Lbs.*
52	Indian Territory No. 4	Nut, bright	45.33	35.44	6.24	12.99	3.86	10 22	9,296	186.1	21.04	7 53	3.76	4.64
45	Iowa No. 1	Nut, dull	39.89	33.08	8 69	18.34	6 39	10.02	10,331	197.7	23.23	7.24	3 91	4.82
67	Iowa No. 2	Nut, very dirty	33.73	35.35	14 88	16.04	4.73	9.92	10,986	192.7	23.28	7.05	4.01	4 95
49	Iowa No. 3	Nut, dull	35.77	36.14	12.44	15.65	6.07	10.03	10,668	189.3	22.96	7.02	4.03	4 97
47	Iowa No. 4	Mine run, dull	37 28	34 09	13 48	15.15	5.04	10.00	9,385	167.3	20.02	7.11	3.98	4.91
66	do	Large briquettes	37 85	36.50	13.24	12 41	3.90	10.03	9,900	184.5	21.11	7 43	3.80	4.70
55	Iowa No 5	Nut, medium bright	38.83	31.76	16 01	13.40	3.09	9.98	11,200	204.7	23.23	7.50	3-77	4 66
6	Kansas No. 1	Lump, dull	49.46	33.78	5.90	10.86	3.82	10.10	7,222	158 2	16 60	8.11	3.49	4.31
7	do	Mine run, dull	48.57	32.68	4.80	13 95	4.94	9 70	7,492	184.0	18.13	8.63	3.28	4.05
5	Kansas No. 2	do	46.68	31.23	4 18	17.91	6 27	7.00	5,600	181.5	18.90	8.17	3.46	4.28
13	do	Nut, dull, washed	51.22	34.32	5.82	8.64	3.77	9.38	8,111	209 8	20.08	8.89	3.18	3.93
3	Kansas No. 3	Mine run, dull	50.99	33.52	2 03	13.46	5.39	9.91	7,700	169.6	18.77	7.69	3.68	4 55
4	do	do	51 05	34 30	2.25	12.40	4 80	10 40	7,101	148 8	16.45	7.69	3.68	4.55
58	Kansas No. 4	Nut, bright	43.59	36.32	5.51	14.58	8.46	10.05	7,835	165.5	18.18	7.75	3.65	4.51
72	Kansas No. 5	do	51.36	32.42	4.31	11.91	4.40	9 90	8,651	215.2	20.62	8.88	3.18	3-93
60	Kentucky No. 1	Nut, very clean	55.59	35.61	2.89	5.91	1.19	10.03	8,355	212.3	19.95	9.06	3.12	3.86
57	Kentucky No. 2	Nut, bright	45.75	37.91	7.76	8.58	3-37	10.00	9,557	208.5	21.75	8.16	3-47	4.28
76	do	Large briquettes	44.32	37.07	7 11	11.50	3.71	9.82	9,371	207.1	21.87	8.06	3.51	4-33
64	Kentucky No. 3	Nut, bright	44.84	37.32	7.92	9.92	3.91	10.07	9,645	211.5	21.75	8.27	3.42	4.22
62	Kentucky No. 4	do	45 74	35.65	5.89	12 72	3 72	9.93	9,386	211.7	21.90	8.21	3.44	4.25
12	Missouri No. 1	Mine run, dull	40.64	34.88	7.28	17 20	4.37	10.00	9,737	207 4	22.30	7.92	3.57	4.41
23	do	Large briquettes	41.85	37.60	6.38	14.17	4.56	5.10	4,500	191.2	20.37	7.99	3.54	4.37
15	do	Nut, dull, washed	44.21	36 81	7.93	11.05	3.58	9.91	9,041	207.0	20 71	8.50	3.33	4.11
37	Missouri No. 2	Nut, dull	37.33	32 88	13.09	16.70	4.92	9 98	11,650	208.2	25 00	7.08	3.99	4.93
44	do	do	39 76	31.77	11.57	16.90	4 60	9 98	9,248	172.5	20.20	7.26	3.90	4.81

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78	Missouri No. 3	Nut, dull	29.98	26.18	18.63	25.21	3.85	9.95	10,828	149 5	21.85	5.82	4 86	6.00
77	do	Nut, dull, washed	39.61	31.18	20.78	8.43	2.88	9.93	11,044	189.7	21.72	7.43	3.81	4.70
70	Missouri No. 4	Nut, dull	42.11	40.10	12.24	5.55	4.98	9.98	9,515	214.1	20.64	8.83	3.20	3.96
27	New Mexico No. 1	Mine run, clean	41.57	37.85	11 90	8.68	.56	10.05	10,963	206 3	23.70	7.41	3.96	4.90
26	New Mexico No. 2	Pea, dull	36.11	37.30	9.92	16.67	1.13	10.00	11,870	198.5	26.37	6.41	4.41	5.45
30	do	Small briquettes	39.07	37.56	6.75	16.62	1.61	8.12	7,765	185.8	22.00	7.19	3.93	4.86
33	North Dak. No. 1, lignite	Nut, brown	25 40	28.13	35.84	10.63	1.68	9.85	10,680	108.9	17.15	5.40	5.24	6.47
1	Pennsylvania No. 1	Mine run, bright	75.69	15.80	1.10	7.41	1.49	9.88	6,365	166.9	15.70	9.04	3.11	3.84
2	Pennsylvania No 2	do	76.76	16.61	.59	6.04	.91	10.08	6,835	191.2	16.60	9.79	2.89	3.57
36	Pennsylvania No. 3	Small briquettes	55.00	27.62	3.00	14.38	1.13	10.08	8,012	184.5	19.01	8.26	3.42	4.23
24	West Virginia No. 1	Nut, bright	56.25	34.64	1.90	7.21	.98	9.98	7,818	199.2	18.94	8.95	3.16	3.90
25	West Virginia No. 2	Mine run, bright	48.80	39.23	2.01	9.96	2.71	10.22	8,354	212.4	19.75	9.14	3.09	3.82
28	West Virginia No. 3	Nut, bright	56.11	30.31	2.54	11.04	1.38	9.97	8,225	211.6	19.82	9.08	3 11	3.84
31	West Virginia No. 4	do	59.84	27.64	2.53	9.99	.96	10.00	7,895	215.1	18.98	9.65	2.93	3.62
34	West Virginia No. 5	do	58.66	28.95	2.11	10 28	1.01	10.08	7,700	207.8	18.44	9.59	2.95	3.81
39	West Virginia No. 6	do	70.03	22.38	2.14	5.45	.70	9.95	7,406	213.2	17.95	10.09	2.80	3.46
43	do	do	71.42	21.44	2.11	5.03	.64	10.18	7,264	208.5	17.21	10.30	2.74	3.39
51	West Virginia No. 7	do	68.27	20.23	2.68	8.82	1.52	10.18	7,700	210.1	18.15	9.85	2.87	3.55
53	West Virginia No. 8	do	56.68	31.19	5.26	6.87	.74	9.98	7,997	211.6	18.72	9.62	2.94	3.63
69	West Virginia No 9	do	59.47	31.11	3.42	6.00	.82	10.00	7,464	210.8	17.78	10.09	2.80	3.46
54	West Virginia No. 10	do	73.84	18.23	1.74	6.19	.59	9.93	7,441	205.9	18.15	9.65	2.93	3.62
56	West Virginia No. 11	do	68.36	16.31	4.85	10.48	.47	9.97	7,700	213.7	18.13	10.03	2.82	3.48
46	West Virginia No. 12	do	75.33	18.26	1.58	4.83	.59	10.13	7,399	206.1	17.68	9.90	2.86	3.53
59	do	Small briquettes	67.46	24.02	2.32	6.20	.84	10.25	7,515	211.8	17.66	10.20	2.77	3.42
63	Wyoming No. 1	Nut, bright	31.61	40.56	21.81	6.02	.63	10.15	11,946	186.6	22.69	7.00	4.04	4.99
61	Wooming No. 2	Nut, very dirty	34 58	35.55	11.10	18.77	3.87	9.95	12,028	184.3	26.51	5.92	4.77	5.90

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West Virginia No. 6. First test: Run-of-mine coal from mine of the New River Smokeless Coal Company, Rushrun, W. Va.

Second test: Same as above.

West Virginia No. 7. Run-of-mine coal from mine of the New River Smokeless Coal Company, Sun, W. Va.

West Virginia No. 8. Run-of-mine coal from mine of the Gauley Mountain Coal Company, Ansted, W. Va.

West Virginia No. 9. Run-of-mine coal from Vulcan mine of the Mount Carbon Coal Company, Limited, Powellton, W. Va.

West Virginia No. 10. Lump and run-of-mine coal from Stuart M. Buck, Mora, W. Va.

West Virginia No. 11. Run-of-mine coal from mines Nos. 1 and 2 from W. H. Coffman, Zenith, Va.

West Virginia No. 12. First test: Run-of-mine coal from mine of the Big Sandy Coal and Coke Company, Big Sandy, W. Va.

Second test: Small briquettes made with patent binder.

Wyoming No. 1. Black lignite from mine of the Wyoming Coal and Mining Company, Monarch, Wyo.

Wyoming No. 2. Run-of-mine coal from Antelope Nos. 1 and 2 and Jumbo mines of the Cambria Fuel Company, Cambria, Wyo.

RESULTS OF THE COAL TESTS UNDER THE BOILERS.

The accompanying table shows some of the most important practical results of the tests of coals burned under the boilers. In each test an effort was made to operate the boiler at a point very near its rated capacity. In some cases the coals tested presented difficulties which made it impossible to accomplish this result. The various coals were tested in the order of the "test numbers" given in column 1, and tests of coals from widely separated localities were frequently made on successive days.

Attention is called to "rate of combustion" at which the coals were burned. To those familiar with tests of this char-

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acter it will be evident that the coals were for the most part burned at a rate calculated to give approximately the most favorable results.

All of the chemical results reported in this table were furnished by the chemical laboratory in charge of Prof. N. W. Lord, which is sufficient guarantee of the accuracy.

The "horsepower developed by the boiler," recorded in column 11, refers to the standard boiler horsepower and is the evaporation of 34.5 pounds of water per hour from a feed-water temperature of 212 degrees Fahrenheit and at atmospheric pressure.

In column 13 will be found recorded the number of pounds of water evaporated by one pound of dry coal at and from a temperature of 212 degrees Fahrenheit. This column gives the best comparative results of the relative values of the coals tested, as far as these results relate to their commercial values.

The final report on these tests will give the values of all the items of the standard code report of the American Society of Mechanical Engineers and the complete logs of all tests, as well as a graphic chart of each trial.

PRACTICAL DATA ON EUROPEAN PRACTICE WITH STEAM TURBINES.

The remarkable development of the steam turbine in this country in connection with power plants makes it of interest to discuss European practice, which has covered a period beginning, perhaps, ten years before active work was started here. The lowest steam consumption under actual working conditions in Europe is 9.11 pounds per indicated horsepower hour, although this does not include the power required to operate the condenser outfit, which in this country is usually considered not over 5 per cent. The lowest record for reciprocating engines is 9.41 pounds (including condensers), which is attained both at one of the street-railway power plants in Berlin and in Vienna.

Although there are some half-dozen different types of turbine

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in use on the continent, the Brown-Boveri-Parsons is the most popular, being installed in some 150 plants throughout Europe.

The rapid increase in the use of the turbine during the past five years was due very largely to the successful operation of the municipal plant in Elberfeld, Germany, for running the suspended railway between Elberfeld and Barmen. Here they have two 1,000-kw. Parsons turbines, which operated, even at the early date of their installation, with a minimum steam consumption of 12.3 pounds per indicated horsepower per hour for one, and 12.1 pounds for the other.

The steam turbine, like the reciprocating engine, may be run both condensing and with free exhaust, and the jet condenser, the surface condenser, the siphon condenser or a central condensation system may be employed. As there is no oil used in the steam chambers of the turbine, the water from the condenser may be used directly for boiler feed or for other purposes, and it is not necessary to filter it and thus lose a large portion of its heat. The absence of oil in the condensed steam is of special advantage when surface condensers are used, as in this case the cooling water does not mingle with the condensed steam, and the feed water returned to the boilers is thus always pure and free from sediment. It is for this reason that surface condensers are specially appropriate for turbine plants. Quite contrary to the practice in this country, most of the condensers and pumps in these plants in Europe are driven by electric motors.

A turbine lends itself much more readily to the advantageous use of superheated steam than a reciprocating engine, on account of the absence of pistons, stuffing boxes and cylinder lubrication. As the economy due to superheated steam increases directly with the temperature, the turbine presents another advantage, because there is practically no limit to the superheat it can stand.

The steam consumption of the Parsons turbine, used on the continent, as already shown, will compare favorably with that of the best compound engine built today. The accompanying table gives some data regarding the steam consumption of

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Parsons turbines furnished by Brown, Boveri & Co., of Baden, Switzerland, and installed in a number of different plants. All the turbines are run condensing, and the steam consumption given in the table includes that required for the auxiliaries, except those cases marked*, where this work is not included. The table shows clearly the decrease in steam consumption as the size of the turbine and degree of superheat increase, and from the data here given it may safely be expected that in the 10,000-H.P. turbine, which is at present being constructed by Brown, Boveri & Co., that the steam consumption will be considerably less than 9 pounds per indicated horsepower per hour.

In comparing the results of the turbine tests with those of reciprocating engines it must be borne in mind that it is usual to make special preparations before testing reciprocating engines, while there being no parts of the turbine subject to excessive wear, there is no necessity for special preparation at the time of test; indeed, tests made several years apart will show the same results.

TABLE SHOWING PERFORMANCES OF BROWN-BOVERI-PARSONS TURBINES, DIRECT-CONNECTED TO C. E. L. BROWN ELECTRIC GENERATORS.

Location of plant.	Output.	Output.	Steam pressure, absolute.	Temperature, steam.	Steam per I.H.P. hour.
	kw.	*H.P.*	*pounds.*	*° F.*	*pounds.*
Refrigerating plant, Berlin*	100	133	129	372	15.6
Electric plant, Chur	200	267	184	482	13.6
French navy yard*	280	420	216	†	13.8
Nord-Deutcher Lloyd, Bremen	300	400	147	†	14.
Steam railway, Linz, Switzerland	300	400	132	†	14.3
Cellulose factory, Villach	350	467	169	482	12.1
Rolling mill, Antonienhütte*	400	533	110	†	13.
Ship yard, Kiel*	400	533	132	†	13.
Spinning mill, Colmar*	400	533	162	445	11.7
Mine and pottery, Tschopeln	400	533	110	406	13.
Iron and steel works, Diedenhofen	450	600	118	482	11.8
Iron and steel works, Hösch*	500	667	110	452	12.5
Schlieper & Baum, Elberfeld	500	667	147	482	11 6
Municipal plant, Frankfort*	3,000	4,000	162	572	9.11

* Steam for power to operate condenser not included. † Saturated.

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As an example of the stability and slight wear of the turbine, it may be interesting to refer to the 500-H.P. Brown-Boveri-Parsons turbine in the Carlshuette at Diedenhofen, Germany, which was in service 5,800 hours continuously, except for a single stop of ten minutes' duration. At the end of the run the turbine was taken apart by the company's engineers and examined carefully for evidence of wear, of which not the slightest could be found.

In comparing performances of reciprocating engines with those of turbines, it is necessary not only to take into consideration the steam consumption of the engine and turbine, but also to consider the wear and tear on the engine and the oil required.

The Bavarian Inspection Society thoroughly investigated this subject of oil, and found that in compound and triplex expansion engines of 100 to 1,500 H.P. the oil consumption was about 2 grammes per horsepower hour, costing 1/40 cent. In the Parsons turbine the oil consumption is practically negligible as compared with reciprocating engines, the figures being 0.1 to 0.3 gramme per horsepower hour in units of 100 to 1,500 H.P. With the reciprocating engine the cost of the oil consumed amounts to 7 to 15 per cent. of the cost of the coal used, while with the Parsons turbine the cost of the oil ranges from ½ of 1 per cent. to 2 per cent. of the cost of the coal.

The attendant delay due to the preparation of an additional reciprocating unit for service is entirely avoided with the turbine, for it is equipped with a by-pass valve, by means of which steam at full pressure may be admitted to the second or low-pressure steam chamber, and the capacity of the turbine is thus largely increased. This by-pass is also a great advantage in case of an accident to the condensing apparatus which would necessarily decrease the power of the turbine, just as it would in the case of an engine. The use of this by-pass, however, means a less economical steam consumption.—Franz Koester, in "Engineering Record."

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VARIATIONS IN ANGULAR VELOCITY OF SHAFTING.

The following notes, published in "London Engineering," are from a paper read by Mr. J. H. Heck, before the Institution of Naval Architects:

Mr. Heck had devised an instrument by which the angular velocity of the tunnel shaft could be graphically recorded during parts of a revolution. It consisted of a small electric motor, which was kept revolving at a high speed, current being supplied by a storage battery; this was mounted above the shaft to be operated upon. At the lower end of the spindle of the motor there was a brass sleeve, fitted eccentrically, carrying a pencil holder, so that when the spindle was revolved and a sheet of paper was held stationary underneath, small circles of $\frac{5}{16}$-inch in diameter would be traced. The motor was fixed to a small teak frame, which could slide easily along a teak batten or straight-edge about 6 feet to 7 feet long, fixed in the shaft tunnel in a fore-and-aft direction, the position being such that the center of the spindle of the motor was directly over the longitudinal center line of the shafting. The pencil was kept in contact with the paper by an elastic spring. It will be seen that a sheet of paper placed around the shafting would have traced on it by the pencil a loop-curve. When the engines were running at various speeds, the curves were obtained by bringing the pencil in contact with the paper and starting the motor, which, when in full motion, was pushed slowly along the batten by hand, the pencil thus describing what the author called a continuous screw-thread of loop curves which enveloped the shafting. The distance between each small curve from crest to crest represented the distance passed over by a point in the periphery of the shaft during a small uniform interval. By counting the number of loops, the number of revolutions of the motor for one revolution of the engine was determined, and by measuring the distance between the crests of the small curves or loops and dividing this distance by the radius of the shaft, the angle in radians passed through by the crank could be arrived at.

Some interesting results were obtained in this way. In a

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cargo vessel fitted with two-crank compound engines, the propeller well immersed, on comparing one-eighth of a revolution with another one-eighth of the same revolution, the variation in the angular velocity of the shafting at 58 revolutions was 12 per cent.; making the same comparison between two quarter revolutions the variation was 8.6 per cent. With a small vessel with two-crank compound engines working at high pressure, the propeller immersed, one-eighth of a revolution compared with another one-eighth of the same revolution gave a variation in angular velocity of the shaft at 93 revolutions of 12 per cent., and at 60 revolutions 24 per cent. Making the same comparison with quarter revolutions at 93 revolutions, the variation was 9.8 per cent. A large cargo and passenger steamer, fitted with three-crank triple-expansion engines, working at 180 pounds pressure, propeller just immersed, gave, on comparing one-eighth of a revolution with another one-eighth of the same revolution, a variation in angular velocity at 66 revolutions of 4.6 per cent., and at 54 revolutions 5.5 per cent. This was a good result. Making the same comparison with quarter revolutions at 66 revolutions, the variation was 2.1 per cent. With a cargo steamer fitted with three-crank triple-expansion engines working at 180 pounds pressure, the tips of the propeller blades 20 inches out of the water, a comparison of one-eighth of a revolution with another one-eighth of the same revolution gave a variation of angular velocity of the shafting at 71 revolutions of 5.6 per cent., and at 61 revolutions 4.6 per cent. Making the same comparison with two quarters of a revolution, the variation at 71 revolutions was 2.15 per cent.

With a large cargo vessel, fitted with a four-crank quadruple-expansion engine, balanced on the Yarrow-Schlick-Tweedy system, working at 110 pounds pressure, tips of the propeller blades 55 inches out of the water, on comparing one-eighth of a revolution with another one-eighth of the same revolution the variation in the angular velocity of the shafting was, at 66 revolutions, 4.6 per cent.; at 48 revolutions, 4.4 per cent.; comparing two quarter revolutions at 66 revolutions the variation

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was 1.2 per cent. In the case of a small high-speed passenger engine, fitted with a four-crank quadruple-expansion engine, balanced on the Yarrow-Schlick-Tweedy system, and working at 200 pounds pressure, the propeller being immersed 4 inches, it was found, on comparing one-eighth of a revolution with another one-eighth of the same revolution, that the variation of the angular velocity of the shafting at 104 revolutions was 4.9 per cent. and at 55 revolutions 5.5 per cent. This was also a good result.

As an illustration of the effect of slight racing on the variation of the angular velocity of shafting a case was taken of a large cargo and passenger steamer, a sister vessel to the one above named, fitted with three-crank triple-expansion engines. The propeller was fully immersed, the tips of the blades being 24 inches under water. The engine was racing at times, though not very much. When no racing was apparent, on comparing one-eighth of a revolution with another one-eighth of the same revolution, the average variation of velocity in the shafting at 68 revolutions was 7 per cent. to 9 per cent.; while when racing was evident, comparing one-eighteenth of a revolution with another one-eighteenth of the same revolution, the variation in velocity was 21 per cent.; or, in other words, while the average number of revolutions of the engine was 68 per minute, the engines actually during part of one revolution were going at the rate of 82 revolutions, while during another part of the revolution they were going at the rate of 58 revolutions per minute. It was to be judged from this that, if engines were racing heavily at sea, it would be found that the acceleration at intervals in the rate of revolution per minute would be considerable, and quite sufficient to account for the signs of straining which had appeared in the machinery of vessels that had encountered very bad weather. At low speeds of revolution the variation in the angular velocity per revolution was nearly always greatest, and as the revolutions increased in number the variations became less. When there was an exception to this rule vibration became apparent, and when vibration increased the variation in the angular velocity increased.

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In the discussion which followed, Mr. Cole asked what precautions were taken to ensure uniformity of speed of motor.

Mr. R. E. Froude thought it was certainly an important question whether engines went round at a uniform speed. One interesting point that came out of this investigation referred to the slip of the propeller. Variation in the speed of revolution would naturally cause additional strains upon the propeller blades.

Mr. Heck, in replying to the discussion, said that the paper gave the bare results. The Germans had done much in investigations of this nature, and he was glad to follow their lead. The question was one which would have to engage the attention of engineers. The trials showed that when the variation in angular velocity of the shafting was least the consumption of coal was most favorable. Vessels in heavy weather must have an acceleration that was something enormous, and which had not been considered even by the best engineers. It was known how any movement, such as chattering, absorbed power in any machine. Another point that came forward was that the heavy engine gave less variation than a light one. The motor was so wired that the speed of rotation was constant. Nothing, of course, in this respect was perfect, but he thought they had arrived at conditions that were not far out. He had used more than one motor, and the results with different machines practically confirmed each other.

WATER-TUBE BOILER DESIGN.

Water-tube boilers may be broadly divided into two classes, according to the general position of the tubes. In one class the tubes run in a vertical, or nearly vertical position, while in the other the position is nearly horizontal. The respective merits of these two arrangements are discussed at length in a paper presented before the North-East Coast Institution of Shipbuilders by Mr. W. R. Cummins, and the general subject of the design of water-tube boilers is treated at length in the same connection.

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Mr. Cummins naturally considers the boiler with especial reference to marine service, but the general points apply to stationary purposes as well. The conditions necessary for commercial success in a boiler are enumerated as those affecting the fuel economy, the weight and space economy, the cost or capital economy, and the economy of operation or maintenance. The relative importance of these various conditions depends somewhat upon the service, the matter of weight, for instance, being an important question for a marine boiler, and of secondary consideration for land service.

The conditions necessary for high fuel economy are stated to be perfect combustion and perfect efficiency of heating surface.

"Perfect combustion is an unmixed blessing and does not interfere with any of the other economical conditions. On the other hand, to secure perfect efficiency of the heating surface would mean an almost unlimited amount of it, which would seriously interfere with weight and cost of economy.

"One of the chief sources of loss in burning fuel in a boiler is due to the excess air which has to be admitted to provide for complete combustion to CO_2 and H_2O. The whole of this excess air has to be heated up to the temperature of the rest of the gases and thus carries off a great deal of heat to the funnel. This source of loss is common to all boilers and may be largely reduced by skilled firing and proper proportioning of firebar spaces and air admission. It is very probable that with oil fuel the excess of air over that actually required for perfect combustion is not so much as with coal fuel; hence the furnace efficiency should be greater."

In considering steam-boiler economy it must always be remembered that the furnace is distinctly a piece of chemical apparatus, and that its function is the production of heat by the combustion of fuel, while the boiler proper is a piece of physical apparatus, designed to absorb the heat produced in the furnace and convert it into mechanical energy in the steam.

Considering the furnace in itself, it is generally admitted

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that the form and size of the combustion chamber is an important element. By the combustion chamber is meant the space immediately succeeding the grate, along the direction of the movement of the products before the actual heating surface of the boiler is reached. In order that the combustion should take place effectively it is necessary that the gases should be caused to mingle thoroughly with the air, and that sufficient time should be given for the chemical changes demanding a high temperature to be completed before the cooling surfaces of the boiler are reached. The rate of transfer of the heat from the gases to the water depends upon their difference in temperature, so that the efficiency of both the furnace and the boiler depend to a large degree upon the production and maintenance of a high temperature in the combustion chamber. In any case the time occupied by the passage of the gases through the boiler is never more than a few seconds, so that every facility for the completion of the combustion and the transfer of the heat must be given if even a fair degree of efficiency is to be secured.

Mr. Cummins points out that the various designs of boilers with vertical water tubes give an excellent opportunity for the provision of a large combustion space having its greatest dimension in the line of the flow of the gases, while those forms in which the tubes are horizontal, or nearly so, have the lower rows of tubes just over the grate bars, where the green gases coming off the fuel must be cooled down below combination temperature. In some forms of boilers the combustion takes place in a sort of Dutch oven, placed in front of the boiler proper, this being a recognition of the desirability of providing for better facilities for combustion before the heating and cooling surfaces of the boiler are reached.

Coming to the elements in design which affect the efficiency of the boiler itself, Mr. Cummins shows that the maximum possible economy in the transfer of the heat depends upon heating-surface efficiency. Theoretically the greatest efficiency would be obtained when the temperature of the escaping gases in the chimney was just equal to the tempera-

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ture of the water in the boiler, but such a condition is commercially impracticable. In actual practice the attention which can be given to heating-surface efficiency is limited by other requirements, and, like all other engineering structures, a steam boiler is a compromise.

When economy in weight is the controlling question the heating surfaces are necessarily restricted, and the chimney temperatures are consequently high. In like manner, economy in cost involves limitations to heating surface and corresponding diminution in efficiency. In every case, however, it is desirable to put the heating surface into the most efficient condition, and the elements by which the designer is able to influence the quality of heating surface are enumerated as thickness of transmitting surfaces, circulation of the water and circulation of the gases.

The thickness of the surface is not a matter of very great influence upon the transmission of heat, within the limits of thickness ordinarily used in steam boilers, the condition of the surfaces being more important, and freedom from soot on one side and scale on the other are of greater moment, so that facility for keeping both surfaces clean enter into the design.

So far as circulation of the water is concerned, the vertical tubes undoubtedly have the advantage, and none of the devices used in horizontal tubes, such as inner tubes or diaphragms, circulating nozzles, etc., need be employed when the tubes are placed nearly in the vertical position. Boilers using vertical tubes are also free, in most cases, from the use of narrow and tortuous headers, so that a freer entrance and discharge may be provided.

Considering the circulation of the gases, it has already been noted that the entire duration of contact is very short, but it should be prolonged as much as possible, since the time element is an influential one. By the introduction of suitable baffle plates the gases may be prevented from passing directly to the chimney, and directed against portions of the heating surface which would otherwise receive less heat. In this respect there is not much difference between vertical and horizontal tubes,

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examples of both kinds being cited in which the baffles are well arranged to prolong the duration of contact. An important matter in connection with baffle systems is their maintenance, since the presence of leaks will materially diminish their effectiveness, and in this respect the vertical tubes have an advantage.

Mr. Cummins considers that it is best to have the path of the gases as nearly at right angles to the tubes as possible, and this idea seems to be generally prevalent. It has been pointed out, however, that the direction of contact of the gases with the tubes is not a matter of great importance, since the real cause of the transfer of the heat is the difference of temperature and not the "impinging" of the gases against the metal. It is well known that vertical fire-tube boilers are fully as efficient as the horizontal variety, and the same should be true in water-tube boilers regardless of the direction of the transfer of the heat.

Summing up, Mr. Cummins says, that so far as fuel economy is concerned, a large and high combustion chamber is an absolute necessity; and that vertical tubes, as small in diameter as practicable, are preferable in all respects to horizontal ones. It is equally important that the gases, on leaving the combustion chamber to enter among the tubes, are not rushed through too quickly, but are allowed sufficient time to give up their heat. It is also desirable that the gases be split up and divided into as many narrow streams as possible.

Mr. Cummins considers at length the conditions involved in the use of water-tube boilers on shipboard, especially in connection with economy in weight, and he shows the advantages of the vertical tubes in this respect also, his views in this direction agreeing closely with those of the Admiralty boiler committee.—"The Engineering Magazine."

SUBMARINES.

Sir William White, former Chief Naval Constructor of the British Navy, an expert of international reputation, who re-

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cently came to America to make a study of the condition, tendency and policy of the United States Navy, has written three papers on submarine boats for the "London Times," which are of large practical interest to naval experts everywhere. The first of these articles follows:

The construction of submarines for the Royal Navy was commenced in 1900. Five vessels were then ordered of Messrs. Vickers, Maxim & Co., who had secured from the Holland Company, of the United States, the right of building vessels similar to those which had been recently ordered for the American Navy. Mr. Holland had devoted himself to the construction of submarine vessels for more than twenty years, and this continuous effort, combined with great skill as a designer, had produced gradual improvement and eventual success. When the Navy Department invited competitive designs for submarines, it was natural, therefore, that Mr. Holland should prove successful. The construction and experimental trials of his vessel occupied a considerable time; but the Naval Commission which conducted the test experiments finally pronounced in favor of the type. Before this result was reached a new company had been formed to develop the Holland patents, and it was this company which first approached the Admiralty and subsequently arranged with Messrs. Vickers to work their patents.

The Admiralty thus insured success in preliminary orders for submarine vessels, because they acquired the accumulated experience of Mr. Holland himself, with a guarantee that further improvements made in the United States during the continuance of the agreement should be placed at their disposal through Messrs. Vickers, and the resources of that firm (whose predecessors at Barrow had built a number of submarines of the Nordenfelt type) were also secured. In addition it was decided by the Admiralty to detail an able naval officer of large experience in electrical and torpedo work to watch the construction and conduct the trial of vessels built by Messrs. Vickers, while the experience of the constructive and egineering departments at the Admiralty was made available. The

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strongest possible combination was thus arranged, of all classes whose co-operation tended to produce satisfactory results in the construction, equipment and trials of the new class of vessel.

These arrangements differed essentially from those which had been adopted in France and are still employed. The French system gives free scope to the ideas and abilities of rival designers in the Génie Maritime, some of whom have individually produced new designs before the trials of their original designs were completed. It may be thought that by thus bringing many competitors into the field superior results would be obtained, but this hardly appears to have been true; nothing like an approved type has been established as yet in France, most probably because there has not been persistence in experiment and gradual improvement with any class. As the course taken by the Admiralty was in accordance with my advice, it may be thought that I can hardly deal with the subject impartially; but that there are many advantages in the Admiralty method is unquestionable, and the wisdom of the course taken in 1900 has been confirmed by subsequent experience. Delays and difficulties similar to those which have occurred abroad have been avoided, and, in a comparatively short time, a considerable number of submarines have been built and great and successive improvements have been effected, while as a group British submarines undoubtedly compare favorably with any corresponding group of foreign submarines.

It is the fashion to criticise the Admiralty for long delay before beginning the construction of submarines. This criticism when analyzed will be found to be based upon misconception or incomplete consideration of the facts. No doubt French designers have shown remarkable ability in dealing with many naval problems, and they took the lead in this branch of construction largely because the conditions under which they work give greater freedom in making new departures. That lead, however, was due to special circumstances which may be noted in passing. The construction of submarines in France proceeded in a dilatory and half-hearted fashion for several years after the *Gustave Zedé* was ordered in

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1888. Her trials extended over nearly eight years and led to many modifications in details and equipment. When competitive designs for submarines were invited for the Ministry of Marine in 1896, there was no urgency or eagerness in taking action thereupon; in fact, small progress was made until an impetus to more active exertion was given by the Fashoda incident.

The French people then had pressed upon their attention the relative weakness of their Navy, and it was natural that consolation should be found in the distinct lead that they had undoubtedly obtained in submarines as the result of numerous experiments. A type had been produced which was superior to all its predecessors, and could be multiplied rapidly at a moderate cost. Twenty or thirty submarines could be produced for the price of a battleship and in much less time. Under these circumstances the cult of the submarine naturally became popular in France, especially amongst the members of the Jeune Ecole who had condemned the continued building of armored "mastodons" and advocated swift commerce-destroying cruisers. The development of submarines, it was argued, made unnecessary the continued construction of costly battleships; and submarines needed only to be supplemented for sea work by swift cruisers. These extreme views, however, did not represent the matured opinion of experienced French naval officers and naval constructors, and they were not acted upon. Many submarines were ordered, but the construction of battleships and armored cruisers went on concurrently, and the wisdom of that policy is now almost universally recognized in France.

Writers in the popular press of this country, seeking for "some new thing" to illustrate the shortsightedness and extreme conservatism of British naval authorities, waxed eloquent in condemnation of the Admiralty for long delay in taking similar action. Underlying most of these criticisms was the assumption—which was itself a fallacy—that, if foreign navies adopted submarines, it followed that the Royal Navy must possess an equal or greater force of such vessels, their numbers

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being governed by the numbers of foreign vessels of similar type. In other words, it was tacitly assumed that submarines must constitute the best and necessary reply to foreign submarines. It may be worth stating, although the fact should be obvious, and the statement involves no breach of official confidence, that everything done in France and the United States from 1895 onwards in connection with submarine construction was thoroughly well known and carefully considered by the Admiralty at the time. There was no difficulty in undertaking here the design or construction of submarines had it been considered desirable to do so, and the work of building could have been done with greater rapidity; but it was decided to await developments elsewhere before making a start.

Experimental submarines had been built and tried in this country in recent years; the trials had been witnessed by officers of the Admiralty, and full information was possessed. The largest, and on the whole the best, of the submarines constructed before the introduction of storage batteries and gas engines were those of the Nordenfelt type built at Barrow-in-Furness in 1886 from designs largely due to Captain Garrett's initiative. When it became possible to use electric propulsion for the submerged condition of these vessels, in combination with gas engines for their surface propulsion, advances could be made on the Nordenfelt type, which had been fitted with steam boilers and reciprocating engines, and consequently ran risks in diving which can be avoided under present conditions. It was no pleasure trip to go under water in a Nordenfelt vessel with boilers hermetically sealed, and propulsive power dependent upon heat stored in boilers and in special water tanks provided for the purpose. But, while improvements in propelling apparatus had considerable effect on the design of submarines, it still remained true that this country had not fallen so much behind France or the United States as was affirmed, in the construction and trial of submarines. These experimental vessels were built by private enterprize, but the results were freely placed at the disposal of

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the Admiralty, and the naval service benefitted by the trials. It was with this knowledge before them that their Lordships took no action up to 1900; being of opinion that the time had not yet come when it was necessary or desirable to commence the construction of submarines. Nor was there in the Navy itself at that date any marked or general desire for submarines; while in other navies, except the French, practically the same policy of waiting was adopted. In the United States a private company, not the Navy Department, pursued the investigation to a successful issue, and official action followed. Even the French did not embark upon a definite program of submarine construction until about six years ago, and the question is admittedly still in an experimental stage.

Many types were under consideration in France when the Admiralty made arrangements with Messrs. Vickers to build five vessels; but only a small number of submarines were ready for service in France at that time, and nearly all of these were avowedly restricted to coast and harbor defense. French official statements recently published show that up to 1897 the *Gymnote* and *Gustave Zedé* stood alone; in 1897–98 two more submarines were ordered, followed in 1899–1900 by ten other vessels of the submarine and submersible types. At the end of 1900 not more than five or six of these had been made available for service. In the United States there were then only two complete submarines, and it is interesting to note that in the United States Navy at the present time there are altogether only eight submarines, although a number of vessels have been constructed there for other navies during the past year. When the Admiralty ordered five submarines of the Holland type it was stated that the main purpose was to obtain experimental information first-hand in order to assess the true value of such vessels in naval warfare and to determine the best means of dealing with their attacks. The latter investigation obviously must have always the greatest importance for the British Navy, since no recognized authority would suggest that the Bristish Navy should depend largely on defensive weapons, or consist chiefly of submarines, even

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if the greatest offensive power which has been claimed for the type were realized in practice. But when other navies laid down many submarines, with the declared intention of employing them for offense and defense, it was absolutely necessary that steps should be taken to ascertain the best means of foiling such attacks, and the only method available was the acquisition and experimental use of similar vessels by the Royal Navy; that plan was adopted by the Admiralty as soon as the threat became serious. Action taken at an earlier date or on other lines would have involved much larger expenditure of time and money in arriving at a decision as to type. French experience, of course, could not be made available, and it was a fortunate circumstance that at the critical moment the Holland Company came forward, placing the British Navy in a position to benefit by their experience and information, and so to start under favorable conditions on a path which has already conducted to a most satisfactory result in relative numbers and efficiency of submarine vessels under the British flag.—"Army and Navy Journal."

ALUMINUM-BRONZE CONDENSER TUBING.

Brass condenser tubing gives so much trouble that anything which will give better service must certainly receive attention. All the various brass mixtures seem to give trouble, and much time and labor have been expended in trying to improve them by the addition of tin, nickel, iron or manganese, but with little result. The verdict in regard to brass condenser tubing seems to be that it is "the nature of the beast."

A number of years ago a series of experiments were carried out upon the manufacture and use of aluminum-bronze condenser tubing, and, while they were not sufficiently extensive to demonstrate beyond doubt that this alloy is, *par excellence*, the material for condenser tubing, the results indicate that question is very promising. The experiments were made with a 4-per-cent. aluminum-bronze (96 per cent. of copper and 4 per cent. of aluminum) and the tubes left in the annealed state.

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From tests made in experimental condensers and upon vessels in actual service it was found that the fault of brass tubing, *i. e.*, disintegration, was absent. It is well known that aluminum-bronze is free from disintegration and crystallization and possesses a more homogeneous structure than found in any other alloy. It is also more non-corrosive than brass.

The difficulties which have been encountered in the production of aluminum-bronze condenser tubing have been only one of manufacture, and when this has been overcome we believe that the use of this alloy for condenser tubing will be extensive.—"The Brass World."

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SHIPS.

UNITED STATES.

Washington—*Launch of.*—The armored cruiser *Washington* was launched on March 18, 1905, from the yard of the New York Shipbuilding Company at Camden, N. J.

The *Washington* is a sister ship to the *Tennessee*, which is being built at the Wm. Cramp & Sons Shipyard in Philadelphia.

General Characteristics.—Displacement, 14,500 tons; length over all, 504 feet 6 inches; length on load-water line, 502 feet; beam, molded, 72 feet 6 inches; draught, 25 feet; speed, 22 knots; with a bunker capacity of 2,000 tons of coal.

Armament.—Four 10-inch breech-loading rifles, 45 calibers in length; sixteen 6-inch rapid-fire guns, 50 calibers in length; twenty-two 3-inch rapid-fire guns; twelve 3-pounder semi-automatics; two 1-pounder automatics; two 1-pounder rapid-fire guns; two 3-inch field pieces; two machine guns of .30 caliber; and six automatic guns of .30 caliber.

Armor.—A belt of armor 7 feet 6 inches wide, 5 inches thick, in wake of the engine and boiler rooms, tapering to 3 inches at the ends, extends the whole length of the vessel on the water line.

Extending vertically from the water-line armor over the main deck space, there will be an armor protection 5 inches thick for a length of 232 feet; at the end of this will be 5-inch transverse bulkheads.

The 10-inch gun turrets will have 9 inches of armor at the front, 7 inches at the sides and 5 inches at the rear, the barbettes being 7 inches at front and 5 inches at rear. The 6-inch guns will be protected by 5-inch armor.

An armored deck 4 inches thick on the sides and 1½ inches on the flat, with a cellulose belt on the side, extends the full length of the vessel.

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There is a 9-inch conning tower and shield and a 5-inch armored tube.

Propelling Machinery.—The vessel will be propelled by two vertical, inverted, direct-acting, four-cylinder, triple-expansion engines with a designed I.H.P. of 23,000, when making 120 revolutions per minute. The cylinder diameters are, H.P., 38½ inches; M.P., 63½ inches; L.P. (two), 74 inches each with a stroke of 48 inches.

Boilers.—The boilers are of the Babcock & Wilcox water-tube marine type, sixteen in number, installed in eight water-tight compartments; they have a total of 1,600 square feet of grate surface and 70,944 square feet of heating surface.

St. Louis—*Launch of.*—The protected cruiser *St. Louis* was successfully launched from the yards of the Neafie & Levy Co., Philadelphia, Pa., on May 6, 1905.

The *St. Louis* belongs to the *Charleston* class. The principal dimensions are: Length on load-water line, 424 feet; extreme breadth, 66 feet; mean draught, at 9,700 tons displacement, 23 feet 6 inches; designed speed, 22 knots.

Armament.—Fourteen 6-inch rapid-fire guns, eighteen 3-inch rapid-fire guns, twelve 3-pounder semi-automatic guns, four 1-pounder automatic guns, eight 1-pounder rapid-fire guns, two 3-inch field guns, two machine guns of .30 caliber and eight automatic guns of .30 caliber.

Armor.—The main side armor, lower casemate, 6-inch gun sponsons and signal tower will be 4 inches thick; conning tower and shield, 5 inches; splinter bulkheads, 2 inches; protective deck, 2½ inches on slope and 1½ inches on the flat.

Propelling Machinery.—The main engines are two in number, of the vertical, inverted-cylinder, direct-acting, four-cylinder, triple-expansion type, with cylinder diameters as follows: H.P., 36 inches; M.P., 59½ inches; L.P. (two), 69 inches. The common stroke is 45 inches.

The engines are designed for 21,000 I.H.P., when making about 133 revolutions per minute, with a steam pressure of 250 pounds at the H.P. cylinder.

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There are two main condensers, each with a cooling surface of about 13,400 square feet.

The boilers are of the Babcock & Wilcox water-tube marine type, sixteen in number, installed in four watertight compartments.

The total grate surface amounts to 1,400 square feet and heating surface to 58,800 square feet.

Minnesota—*Launch of.*—The first-class U. S. battleship *Minnesota*, of some 16,000 tons displacement, was successfully launched from the yards of the Newport News Shipbuilding and Drydock Company, at Newport News, Va., on April 8.

The *Minnesota* belongs to a class of battleships which represents the most advanced ideas in modern naval construction. With her sister ships, the *Vermont* and *Kansas*, which are also in progress of construction, the *Minnesota*, when completed, will be one of the most powerful battleships afloat. The hull of the *Minnesota* is of steel throughout, and fitted with docking and bilge keels. The general dimensions of the vessel are as follows: Length of load-water line, 450 feet; breadth, extreme, at load-water line, 76 feet 10 inches; displacement on trial, not more than 16,000 tons; mean draught to bottom of keel at trial displacement, 24 feet 6 inches; gross draught, full load, about 26 feet 9 inches; total bunker capacity, coal, about 2,200 tons; coal carried on trial, 900 tons; feed water carried on trial, 66 tons; trial speed at sea for four hours, 18 knots.

The main battery consists of four 12-inch breech-loading rifles, eight 8-inch breech-loading rifles, twelve 7-inch breech-loading rifles. The secondary battery is twenty 3-inch (14-pounder) rapid-fire guns, twelve 3-pounder semi-automatic guns, six 1-pounder automatic guns, two 1-pounder semi-automatic guns, two 3-inch field pieces, two machine guns, caliber .30; six automatic guns, caliber .30.

This battery will be mounted as follows: The 12-inch guns in pairs in two electrically-controlled, balanced, elliptical turrets on the center line, one forward and one aft, each with an arc of fire of about 270 degrees. The 8-inch guns in pairs

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in four electrically-controlled, balanced, elliptical turrets, two on each beam, at each end of the superstructure.

The 7-inch guns in broadside on pedestal mounts on the gun deck, behind 7-inch armor, each gun being isolated by splinter bulkheads of nickel-steel of from 1½ to 2 inches thick; forward and after guns arranged to fire right ahead and right astern, respectively; other 7-inch guns to have the usual broadside train.

The guns of the secondary battery in commanding positions, having a large arc of unobstructed fire, and protected wherever practicable.

All the 7-inch guns will be so arranged that their muzzles train inside the line of the side armor, thus leaving a clear and unobstructed side when it is desired to go alongside a pier or vessel.

Arrangements will be made whereby the 3-inch guns on the main deck can be quickly and conveniently dismounted, housed and secured.

The hull will be protected at the water line by a complete belt of armor, 9 feet 3 inches wide, having a uniform thickness of 9 inches for about 285 feet amidships, forward and aft of which points the thickness is gradually decreased to 4 inches at the stem and stern.

The lower casemate armor will extend to the limits of the magazine spaces and reach from the top of the water-line belt to the lower edge of the 7-inch gun ports on the main deck, and be 7 inches in thickness, the athwartship bulkheads at the ends of this casemate being 6 inches thick.

There is a complete protective deck extending from stem to stern, flat amidships, but sloped at the sides throughout, and sloped at each end. It is built up of 20-pound plating throughout, with nickel-steel of 40 pounds on the flat and of 100 pounds on the slopes. Nickel-steel protection is also fitted, hatch covers, gratings, etc.

Cofferdams about 30 inches thick and extending from protective to berth-deck level will be worked from end to end of the vessel, these cofferdams being extended above the berth

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deck, forward and abaft the transverse armor, to a height of about 36 inches.

The cofferdams will be packed with cellulose or other approved water-excluding material.

The magazines and shell rooms are so arranged that about one-half the total supply of ammunition will be carried at each end of the ship. Magazine bulkheads adjacent to heated compartments, such as fire rooms, engine rooms and dynamo rooms, will be arranged with air spaces.

The engines will be of the vertical, twin-screw, four-cylinder, triple-expansion type, of a combined I.H.P. of 16,500. The steam pressure will be 250 pounds. The stroke will be 4 feet. Each engine will be located in a separate watertight compartment.

There will be twelve boilers of the Babcock & Wilcox type, placed in six watertight compartments. They will have at least 1,100 square feet of grate and 46,750 square feet of heating surface, and must be able to furnish steam for the main engines and all the necessary auxiliary machinery and other steam machinery throughout the ship with an average air pressure in the fire rooms of not more than one inch of water.

There will be three funnels, each 100 feet high above the base line.

The vessel has all the up-to-date auxiliary machinery, and electrical devices, voice pipes, telephones, buzzers, gongs, indicators, etc.

There will be a lower bridge both forward and aft, and a flying bridge forward, according to the latest practice. On the flying bridge is fitted a screen of bronze for the protection of the men at the wheel, and there will also be a bronze chart house forward.

There will be steel masts forward and aft, the foremast having an upper and lower top, the mainmast a lower top only. Masts are arranged for wireless telegraphy. There will be one signal yard on each mast; also a searchlight platform forward and aft, with a crow's nest on the foremast.—"Army and Navy Journal."

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Dubuque—*Trial.*—The gunboat *Dubuque* ran her speed trials off Newport, R. I., on May 10 and 11, 1905.

The vessel made eleven runs over the measured-mile course for the purpose of standardizing the propellers on May 10th, and on May 11th the full-speed trial of four hours' duration took place at sea.

The average speed for the four hours was 12.85 knots; the highest speed on the measured mile was 13.41 knots, the engines averaging 238.7 revolutions per minute.

The *Dubuque* and her sister ship, the *Paducah*, were designed for 12 knots. A detailed description of these vessels, together with the report of trials, will appear in a later issue of the JOURNAL.

Submarine Torpedo Boats Nos. 9, 10, 11 and 12—*Contract for.*—On March 6th, the Electric Boat Company of New York secured the contract for submarines *Nos. 9* and *10*, and on March 18th the same company was awarded the contract for *Nos. 11* and *12*, the boats in each case are to be built at the Fore River Ship Yard, Quincy, Mass.

The boats will be of the Holland type, the general characteristics being the same for all; *Nos. 9* and *10* will, however, have a displacement of about 261 tons when submerged, while in the case of *11* and *12* the displacement will be approximately 167 tons. The other dimensions are: Length over all, *Nos. 9* and *10*, 105 feet; *Nos. 11* and *12*, 80 feet 8 inches. Extreme beam: *Nos. 9* and *10*, 13 feet, *Nos. 11* and *12*, 12 feet 4 inches, the boats in each case being designed to withstand a pressure corresponding to submergence of 300 feet.

They are to be completed and ready for delivery within eighteen months from the date of contracts.

In addition to the standardization trials under various conditions they will be subject to the following:

(a) Four hours run in light condition with oil engine only; speed to be developed: *Nos. 9* and *10*, 10½ knots; *Nos. 11* and *12*, 8¾ knots, with an endurance capacity of fifty hours.

(b) One hour at full speed, submerged, with electric motor

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only, one torpedo to be fired from each tube. Speed to be maintained: *Nos. 9* and *10*, 8½ knots; *Nos. 11* and *12*, 8 knots.

(c) A storage-battery endurance trial for three hours in awash condition at maximum capacity of motor, the battery and motor to show a capacity corresponding to a speed of 8 knots per hour in submerged condition for *Nos. 9* and *10*, and 7½ knots for *Nos. 11* and *12*.

The boats will also be required to undergo various trials to determine the time required for submergence, firing of torpedoes and endurance at sea.

Chattanooga—*Forced-Draft Run.*—The following notes are taken from a report of forced-draft run made by the U. S. S. *Chattanooga* during passage from New York to Sanchez, Santo Domingo, with three boilers in use (A, C & D).

In addition to the main engines the following auxiliary machinery was in operation:

2 dynamos.
1 evaporator.
2 main circulating pumps.
2 hot-well pumps.
2 bilge pumps.
1 main feed pump.
2 forced-draft blowers.
1 flushing pump.
Starboard auxiliary condenser.
Ice machine.

At 5·00 A. M. the port forward and starboard after forced-draft blowers were started, their speed being increased to 450 revolutions per minute at 5·30, and at the same time the auxiliary exhaust was turned into the low-pressure receivers.

The starboard counter was out of order during the run, but the engines were kept together by means of the shaft telltales.

No indicator cards were taken during the trial, but the horsepower was taken from cards obtained during the same week as trial.

In addition to the regular watch there were on duty two machinists and also one extra fireman for each boiler; the

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firemen on duty from 6·00 to 8·00 were new men who had never had any experience with forced-draft operation, and it will be noted that during this time the revolutions were from 13 to 14 less than during the last two hours.

New River coal with about 20 per cent. of ash was used. Air pressure in fire rooms maintained at 1½ inches of water.

The bottom was clean, the ship having been out of dock less than one month. The opening of throttles varied from seven-tenths to full.

Cylinder cut-offs were as follows	H.P., .45; M.P., .50; L.P., .55
Average revolutions from 6·00 to 10·00 A. M.	136.7
speed (standardization curve), knots	13.28
(patent log), knots	13.67
Indicated horsepower of main engines	2,145
of auxiliaries	190
total	2,335
Coal per hour, pounds	4,896
I.H.P. per hour, main engines, pounds	2.28
total, pounds	2.10
Distance run per ton of coal, knots	6.07
Pounds of coal per mile	368.7
Draught of ship, forward, feet and inches	15–10
aft, feet and inches	15– 6

The ship was steaming in water with depth of over 2,000 fathoms, with moderate sea, one point abaft port beam, the wind on port beam with a force of 3.

...	Engines.	Revolutions per min.	Steam pressure, in pounds per sq. in.				Steam cut-off, in dec. of stroke from beginning.			Opening of throttle valve, in tenths.	Vacuum in condenser, in inches of mercury.	Pounds of coal per hour.	Temperatures, in degrees Fahrenheit.					Revs. or double strokes per minute.		Air pressure, in inches of water.
			In boilers.	At engines.	In receiver above a per. vacuum.		H.P. cyl.	I.P. cyl.	L.P. cyl.				Engine room.	Fire room.	Injection water.	Discharge water.	Feed water.	Circulating pump.	Fire-room Blowers.	
					1st.	2d.														
A. M.																				
6	Stbd.	...	165	160	28	12	.45	.45	.45	3	27	3,264	90	100	77	87	193	130	450	1½
7		...	155	150	51	17	.45	.45	.45	5	27	3,672	90	110	77	90	140	140	450	1½
8		...	145	140	52	17	.45	.45	.45	7	27.5	2,652	90	110	77	90	138	130	450	1½
		...	195	190	60	24	.45	.55	.60	full	27	6,120	95	110	78	94	174	130	450	1½
10		...	195	190	58	24	.45	.55	.60	full	27	6,120	95	110	78	94	175	130	450	1½
6	Port.	100.5	165	160	53	17	.45	.50	.55	6	27	...	90	100	77	87	193	130	...	...
7		129.7	155	150	53	17	.45	.50	.55	9	27	...	90	110	77	90	140	140	...	...
8		128.5	145	140	53	16	45	.50	.55	7	27	...	90	110	77	90	138	130	...	...
9		144.2	195	190	70	18	45	.50	.55	10	27	...	92	110	78	90	180	130	...	...
10		141.2	195	190	65	17	.45	.50	.55	10	27	...	92	110	78	90	180	130	...	...

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AUSTRO-HUNGARY.

The official trial of the Austro-Hungarian battleship *Erzherzog Karl* took place off Pola on May 3, 1905; 14,000 H.P. was contracted for, and a speed of 19¼ knots. The trial resulted in 16,000 H.P. being developed, and the speed was 20 knots. The machinery for this vessel was constructed by the Stabilimento Tecnico, Trieste, water-tube boilers of the Yarrow large-tube type being adopted.

ENGLAND.

Cochrane—*Launch of.*—On Saturday, May 20th, the armored cruiser *Cochrane* was launched from the works at Glasgow of the Fairfield Shipbuilding and Engineering Company, Limited. The contract was completed in September, 1903, the keel laid on March 24, 1904. The *Cochrane* is of the following dimensions: Length between perpendiculars, 480 feet; breadth, extreme, 73 feet 6 inches; and displacement at load draught, 13,550 tons. There is an armored belt extending all round, consisting of 240-pound armor, tapering to 80 pounds at the fore end and 160 pounds at the aft end of the vessel. The protective deck proper extends from the stem to the stern, and ranges in weight from 80 pounds to 30 pounds per square foot. This deck is worked at the lower edge of the armor, and protects the vitals of the ship. Another deck of 40 pounds per square foot forms a crown over the side armor and armor bulkheads. The normal coal capacity is about 1,000 tons.

The armament will be six 9.2-inch breech-loading guns and four 7.5-inch breech-loading guns, all fitted to revolve in barbettes, the guns being worked and loaded by hydraulic power. The secondary armament consists of twenty-eight 3-pounder quick-firing guns and two 12-pounder guns. The supply of ammunition for these guns will be hoisted by means of electric motors. Two Maxim guns will also be provided. In addition, the vessel has two under-water 18-inch broadside torpedo tubes and one under-water stern 18-inch torpedo tube. At the fore end there is provided a steel conning tower of 10 inches thickness, with an armored communication tube to a lower

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conning tower. Wireless telegraphy on the latest methods will also be supplied. An electric installation is arranged throughout the vessel on the double-wired system, and includes six powerful searchlights.

The propelling machinery consists of two sets of triple-expansion engines, arranged in two watertight compartments. Each set has four inverted cylinders working on four cranks. The crank, thrust and propelling shafting is of forged steel and hollow. The two propellers are of manganese-bronze, and each has three adjustable blades which work outwards when the vessel is going ahead. The main condensers are of oval form, made of gun metal and fitted with brass tubes, and the condensing water is supplied by four centrifugal pumps of gun metal, each fitted with one independent engine. Steam will be supplied by nineteen Yarrow water-tube and six single-ended cylindrical boilers. All are designed to work at a pressure of 210 pounds. The vessel on trial is to develop 23,500 indicated horsepower, and a speed of 23 knots is expected to be obtained. Her complement is about 700 officers and men.

Africa—*Launch of.*—On the same day, May 20th, H. M. S. *Africa* was launched from the Chatham Dockyard. The first keel plate was laid on January 27, 1904. The principal dimenmensions of the ship are: Length between perpendiculars, 425 feet; breadth, extreme, 78 feet; freeboard—at stem, 22 feet; at amidships, 16 feet 6 inches; at stern, 18 feet; draught—forward, 26 feet 3 inches; aft, 27 feet 3 inches. The displacement at load draught is about 16,350 tons. The hull is constructed of steel plates and bars on the bracket system; the stem, sternpost, rudder frame, shaft brackets, &c., being steel castings. On the side, extending from about 5 feet 3 inches below to 8 feet 6 inches above the water-line, the armor consists of Krupp steel, 9 inches and 8 inches thick amidships; 7 inches, 5 inches and 4 inches forward. At the after end of the belt is a rounded armor bulkhead of 12-inch, 10-inch and 8-inch Krupp steel. Above this belt, amidships, is the battery, which is formed of 7-inch Harveyed steel, splinter bulkheads, 2 inches thick, of K.N.C. plating, being fitted be-

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tween 6-inch gun positions. There are also 2-inch nickel-steel plates on bow and stern. The two barbettes for 12-inch guns are circular in front, the upper tiers of plates being 12 inches thick, the lower 8-inch and 6-inch Krupp steel. The conning tower is of 12-inch and 10-inch Krupp steel. The upper deck is formed of two thicknesses of ½-inch plating within the citadel, but only a single thickness of ⅜-inch before and abaft the latter. The main deck has two thicknesses of 1-inch plating from the forward battery bulkhead to about Station 22; from here to stem it is formed of two thicknesses of ¾-inch plating, the remainder being generally one thickness of ⅜-inch plating. The middle deck has two thicknesses of ½-inch plating from the armored bulkhead to the fore side of the forward barbette, where it slopes to the lower deck in two 1-inch plates. The slope on the side over the main machinery compartments is also formed of two 1-inch plates. The lower deck from the slope of the middle deck to the stem is made of two thicknesses of ½-inch plating; and from the armor bulkhead to the after end of the ship, two thicknesses, one of 1-inch and the other of 1½-inch plating.

The armament includes four 12-inch 50-ton breech-loading guns, on two turntables in circular barbettes, with all-round loading. The guns are protected by shields formed of 8-inch and 10-inch Harveyed steel plate, the floor and crown plates being 2-inch and 3-inch armor plating. There are also four 9.2-inch 28-ton breech-loading guns on turntables in circular barbettes, ten 6-inch Mark XI guns in battery on main deck, and twelve 12-pounder 18-cwt. guns—four on upper deck and eight on shelter deck; fourteen 3-pounder Hotchkiss—two on upper deck, eight on 9.2-inch shields and four on forward bridge; two 12-pounder 8-cwt. guns on after shelter deck, and two Maxim guns mounted on tripod stands. There are four submerged tubes for 18-inch torpedoes—two on forward platform deck and two on after platform deck. The ship carries eighteen 18-inch torpedoes and six 14-inch torpedoes, these latter for firing with dropping gear from ship's steamboats. The complement of officers and men is 781. The total capacity

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of coal is 2,200 tons. This amount of coal will enable the ship to steam for about twenty-nine days at a speed of 10 knots, or about four and a half days at 18½ knots.

Messrs. John Brown & Co., Limited, of Clydebank, are the contractors for the machinery. There are eighteen Babcock & Wilcox type boilers, fitted in three boiler rooms, six in each; and three cylindrical boilers, each 15 feet 7⅞ inches external diameter and 10 feet 7 inches long, with three furnaces, 3 feet 11 inches internal diameter, of Brown's cambered type, are fitted in a fourth boiler room. The safety valves to all boilers are to be loaded to 210 pounds per square inch. The engines are of 18,000 indicated horsepower, there being two sets of inverted four-cylinder triple-expansion engines, working at a pressure of 205 pounds per square inch. The diameters of the cylinders are: High-pressure, 38 inches; intermediate pressure, 60 inches; low pressure, 67 inches each. The four main condensers are of brass. The propellers are 16 feet in diameter, ard are arranged so that the pitch may be varied. There are also in the engine rooms four evaporators, four fire and bilge pumps, each capable of discharging 100 tons of water per hour; four main centrifugal pumps, each capable of discharging 1,400 tons of water per hour from the bilge; two twin air pumps driven by independent engines (one in each engine room); two turning engines and two starting engines. In the boiler rooms are eight feed pumps (four main and four auxiliary). Outside the main machinery spaces are four dynamos—two on the lower deck and two on the middle deck, each producing a current of 600 ampères with an electromotive force of 105 volts. There is one ice-making machine in the auxiliary-machinery compartment on lower deck capable of producing 80 pounds of ice in 24 hours under tropical conditions. Four sets of air compressors—two aft and two forward—are fitted on the lower deck for charging and ejecting torpedoes. There is a steam capstan forward, the after capstan being worked by an electric motor.

The steering gear is of the screw type, with two steering engines, one in each engine room. Either engine is of suffi-

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cient power to put rudder from hard over to hard over (or through 70 degrees) in 30 seconds when the ship is proceeding at full speed of 18½ knots. There are four steering positions, viz: fore bridge, conning tower, lower deck forward, and steering compartment on platform deck aft. The 12-inch and 9.2-inch guns are worked by hydraulic machinery, supplied by two pumps. The ship is fitted with six searchlights—two on the masts, two on the forward bridge, and two on the platforms near the after shelter-deck. There are twelve steam fans for ventilating the boiler rooms, twenty-six fans for the general ventilation of the ship, and eight fans for ventilating the engine rooms, driven electrically. The ship has two steel masts, 2 feet 6 inches in diameter, each mast being fitted with a searchlight platform and striking topmasts, each topmast being 146 feet above the waterline, with a multiple-fiber flashing-signal lamp at the fore, and a truck semaphore at the main.

There are sixteen boats provided for the ship: two 56-feet steam pinnaces, one 40-foot steam barge, one 42-foot launch, one 36-foot sailing pinnace, two 34-foot life cutters, one 30-foot cutter, one 32-foot galley, one 30-foot gig, one 28-foot gig, three 27-foot whalers, two 16-foot skiff dinghies. When the boats are armed, the 40-foot steam barge, the 56-foot steam pinnaces and the 36-foot sailing pinnace will each carry one 3-pounder Hotchkiss gun; the 42-foot launch, one 12-pounder 8 hundredweight gun, and each of the 34-foot cutters will carry a 0.303-inch maxim. The steamboats and the 42-foot launch will be fitted with countermining gear, and the 56-foot steam pinnaces with dropping gear for firing 14-inch torpedoes. The weight of hull, &c., when the ship is launched, will be about 5,800 tons.—"Engineering, London."

H. M. S. Forward.—*H. M. S. Forward*, the first of the two scouts built by the Fairfield Company for the British Navy, completed on May 11th her official steam trials, when she realized, as a result of six runs over the measured mile at Skelmorlie, a mean speed of 25.286 knots when developing 14,995 indicated horsepower. The Fairfield Company guaranteed in their contract with the Admiralty a speed of 25 knots, the de-

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signed power being 16,500 indicated horsepower. With a length between perpendiculars of 365 feet and a beam of 38 feet 9 inches, the Fairfield scouts displace 2,850 tons when the draught is 14 feet 2 inches. At this draught 150 tons of coal were in the bunkers, sufficient to carry the vessel 1,500 nautical miles at 10 knots. There is, however, capacity for 500 tons of coal, which gives a radius of action of 5,000 nautical miles at the same speed. The attainment of 25 knots involves a high rate of combustion in the twelve boilers, which are of the modified Thornycroft type. On the official full-power trial the air pressure in the stokehold was under 2 inches, which is moderate for such craft. The boilers, when working under these easy conditions, supplied adequate steam for the engines, which are of the four-cylinder triple-compound type, driving twin screws. The summation of the results, taken each hour on the eight-hours' test, showed that the revolutions averaged 210 per minute, and that the mean power was 15,022 indicated horsepower. The air pressure in the stokehold was 1.8 inch, and the average steam pressure at the boilers 250 pounds, and at the engines 230 pounds. The vacuum was 25 inches. Over the whole eight hours, therefore, the speed was quite 25¼ knots, so that, as already stated, the guaranteed rate was exceeded with a power 1,500 less than was anticipated.—"Practical Engineer."

New Zealand.—The steam trials of the *New Zealand*, battleship, which have just concluded, were very satisfactory. Whilst they were in progress only the usual steaming complement was allowed in the engine room, and the use of water to cool the bearings was dispensed with. The records taken during the 30-hours' trial at a mean collective indicated horsepower of 3,938 were as follows: Draught, 26 feet 3 inches forward, 27 feet 3 inches aft; steam in boilers, 186 pounds; vacuum, 27 starboard, 27 port; revolutions per minute, 70 starboard, 71.1 port; mean total indicated horsepower, 1,936 starboard, 2,002 port; speed, 9 knots; coal consumption, 2.02 pounds per indicated horsepower; total loss of water, 14½ tons. The results of the thirty-hours' trial at 12,918 mean collective indicated

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horsepower were as follows: Draught, 26 feet 3 inches forward, 27 feet 3 inches aft; vacuum, 26.7 starboard, 27.2 port; revolutions per minute, 108.7 starboard, 27.2 port; mean total indicated horsepower, 6,360 starboard, 6,558 port; speed, 16.9 knots; coal consumption, 1.84 pounds per indicated horsepower; total loss of water, 26 tons. Eight-hours' full-power trial: Draught, 26 feet 4 inches starboard, 27 feet 3 inches port; steam in boilers, 197 pounds; vacuum, 25.5 starboard, 26.5 port; revolutions per minute, 120.1 starboard, 121.6 port; mean total indicated horsepewer, 9,084 starboard, 9,299 port; mean collective indicated horsepower, 18,383; speed, 18.59 knots; coal consumption, 2.1 pounds per hour; total loss of water, 12.4 tons.—"Practical Engineer."

King Edward VII—*Steam Trials.*—The *King Edward VII* entered the new *King Edward* dock at Gibraltar on the afternoon of her arrival, a notable event in the annals of the Rock, as she is the first battleship to enter one of the new graving docks. The recent steam trials of the ship supply material for comparison between the water-tube boilers of the ship and the cylindrical boilers with which she is also fitted. At one-fifth of her power the Babcock & Wilcox boilers gave 66.4 revolutions and 3,759 I.H.P., with a coal consumption of 1.74 pounds per I.H.P. per hour. The corresponding figures for the cylindrical boilers were 65.2 revolutions, 3,634 I.H.P. and 1.8 pounds coal consumption. At full power the water-tube boilers gave 198 revolutions, 7,510 I.H.P. and 1.67 pounds coal consumption; while with the cylindrical boilers the readings were 190 revolutions, 6,686 I.H.P. and 1.88 pounds coal consumption. The result of these trials is to show clearly the inferiority of the cylindrical boiler, and will probably prove the death knell of any further attempts to bolster up that now obsolete type of steam generator.—"Exchange."

Garry—*Trial of.*—H. M. torpedo-boat destroyer *Garry*, the last of six destroyers of the new type being built by Messrs. Yarrow & Co., Poplar, for the British Admiralty, was launched on March 21, 1905, and had her full-speed trial on May 2. The result has been awaited with special interest, because the

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shape of this vessel differs considerably from that generally adopted, the after part resembling somewhat the design now common for racing motor boats, the breadth amidships being continued nearly the full width to the stern. The counter thus extends considerably over the propellers, thereby protecting them from damage, and at the same time admitting of more commodious living spaces for the officers without exposing a greater length to gun fire. The rudder, instead of being secured to the stern, as is customary, has its entire surface below the waterline, and therefore is not exposed. The keel is perfectly straight from end to end, the overhang of the stern being of such short dimensions that no special support is necessary, thus simplifying docking operations. The coal-carrying capacity of this form of hull is greater than is usual in the "River" type of destroyer to the extent of 20 tons. These advantages are obtained in combination with a diminished resistance, as the recent trial indicates, the speed attained being 26½ knots, that of other destroyers of the same class varying from 25½ to 26 knots. The dimensions of these vessels are as follows: Length, 225 feet; beam, 23 feet 6 inches. They are fitted with four-cylinder triple-compound engines and four Yarrow boilers. The engines of these vessels are balanced on the Yarrow, Schlick and Tweedy system.

Chelmer—*Trial of.*—H. M. S. *Chelmer* returned to Southampton recently from the Clyde, where she has carried out her official speed and radius-of-action trials. She is the second vessel to complete her trials out of the fifteen similar ships ordered under the Navy Program of 1903–4. The *Chelmer* is a twin-screw torpedo-boat destroyer of the 25½-knot River class. Length, 225 feet; beam, 23 feet 10½ inches; displacement, 550–600 tons; I.H.P., 7,500–8,000. She was ordered from Messrs. John I. Thornycroft & Co., Ltd., on November 24, 1903, and was built at their Chiswick yard, and has now returned to their Southampton works to receive her sea stores and equipment, and for examination before handing over to the naval authorities.—"Page's Weekly."

Boyne—*Trials of.*—Two official trials by the Admiralty of

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the torpedo-boat destroyer *Boyne* have recently been carried out. The first—a full-speed coal-consumption trial—took place on March 10, and the results obtained were so satisfactory as to enable arrangements to be made for the full-power official trial, which took place on March 24th. While running under the Admiralty conditions for the guaranteed speed of 25½ knots she attained an average speed over the four hours of 25.73 knots, thus exceeding her guarantee by nearly a quarter of a knot.

Submarines—*Contracts for.*—Messrs. Vickers have recently received from the Admiralty an order for ten submarines. These boats are to be of a more formidable type, will be more easily submerged and will have greater speed than any built up to the present time.

China Squadron—*Steam Trials.*—The report of the steam trials at sea of the China Squadron is just to hand, and on the eight-hours' run every ship did well, exceeding the official contract speeds, although most of the vessels have been three or four years in commission. The best cruiser speed was got with the *Amphitrite*, built by Vickers, which attained a mean on her eight-hours' sea trial of 21.39 knots, as compared with the designed speed of 20.75 knots. Next comes the *Andromeda*, a dockyard-built ship, engined by Hawthorn, Leslie & Co., which steamed 20.1 knots, only 0.15 miles below the contract rate. Of battleships the *Vengeance*, also a Vickers ship, takes first place, with a speed on the eight-hours' set trial now of 19.10, against the designed speed of 18.25 knots. The other ships are the *Albion*, by the Thames Iron Works, which steamed 18.7 knots; the *Glory*, by Laird, which got 18.6 knots, and the *Ocean*, engined by Hawthorn, Leslie & Co., which got 18.83 knots. The designed speed in these three cases was 18¼ knots. Practically all of these ships have the Belleville boiler.—"The Practical Engineer."

Pathfinder—*Trial of.*—The H. M. S. *Pathfinder*, one of two scouts built by Messrs. Cammell, Laird & Co., Birkenhead, for the British Navy, has now completed her official coal-consumption and speed trials. The coal-consumption trial

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was of ninety-six hours' duration at a cruising speed of about 11 knots. This trial was entirely satisfactory from the point of view of economy. The vessel proved herself capable of a radius of action of about 6,000 knots, which is largely in excess of the requirements of the contract. The eight-hours' full-power trial took place over the measured mile at Skelmorlie on Saturday, April 8th, when a mean speed of 25.48 knots was obtained on the six runs on the mile, and 25.38 knots for the whole time. The guaranteed speed was 25 knots. The vessel proved herself an excellent sea boat.—"Engineering," London.

Submarines—*Launch of.*—On March 8th Messrs. Vickers, Sons & Maxim launched from their yard at Barrow two submarines of the "A" class.

Swale—*Launch of.*—Messrs. Palmer & Company launched on April 10th, from their shipbuilding yard at Jarrow, the torpedo-boat destroyer *Swale*, built for the Admiralty. Her displacement is 565 tons. She will have engines of 7,000 horsepower, and the guaranteed speed is 25½ knots. She is the twenty-fifth destroyer built by the firm for the Admiralty. The first, the *Janus*, was launched eleven years ago. A speed of 27 knots was made by the first three vessels; then came the *Star*, the first of their 30-knot boats. The *Swale* will, like her sister ships, the *Wear* and the *Ure*, now completing, have engines fitted with a system of forced lubrication, and these, with the exception of a destroyer built for the Admiralty some time ago, are the only vessels in the Navy so fitted.—"Practical Engineer."

Spiteful—*Accident to.*—The destroyer *Spiteful* recently, while steaming at a speed of about twenty knots, ran into and practically passed over a barge laden with bags of cement. The bows were stove in, but her light build considerably minimized the force of the shock she gave, as it were, to the impact and was kept afloat until placed in drydock.

A5—*Cause of Accident to.*—The official inquiry held by Captain R. H. S. Bacon and Dr. B. Redwood, F. R. S. E., on behalf of the Admiralty into the explosion on board the sub-

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marine *A5*, which occurred in Queenstown Harbor on the 16th of February, has shown that the accident was due to carelessness and not to any danger that had not up to then been known. The cause of the explosion was shown by the evidence to be due to petrol vapor escaping and becoming ignited by the sparking from the electric motor. A strong smell of petrol was observed some hours before the explosion occurred, caused by the leakage from the petrol pump, due to insufficient packing in the plunger-rod gland. The ventilating fans were started, but apparently the leakage was still going on. The chief engine-room artificer, who had been working at the propeller clutch, was giddy from the smell of the petrol, and had to go forward for fresh air. In spite of this the motors were deliberately started to revolve the engines to exhaust the air, preliminarily to descending below the surface. The sparking of the brushes supplied the necessary ignition of the mixture, and the explosion occurred. The action of starting the motor under such circumstances is described by Captain Bacon as a "flagrant violation of the regulations laid down." It is obvious that in cases where petrol motors are used in confined spaces for driving dynamos, such as motor 'buses and the combined automobile carriages on the North-Eastern Railway, great care must be taken to prevent any leakages of petrol vapor and the consequent forming, on mixing with the air, of a highly explosive mixture, easily ignited by a spark.—"The Practical Engineer."

Destroyer.—It is reported that the British Admiralty intend building a sea-going destroyer, having a displacement of about 700 tons, to be fitted with turbine engines and designed for a speed of 36 knots.

British Shipbuilding.—The following is from the statement of the First Lord of the Admiralty, explanatory of the naval estimates, 1905–1906:

Between 1st April, 1904, and 31st March, 1905, inclusive, the following ships will have been completed and become available for service: Four battleships, *King Edward VII*, *Commonwealth*, *Swiftsure* and *Triumph*; one armored cruiser,

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Cornwall; four third-class cruisers, *Sapphire*, *Diamond*, *Topaze* and *Amethyst;* twelve submarines, nine destroyers, four torpedo boats, one river gunboat and a new Admiralty yacht.

On 1st April, 1905, there will be under construction: Eight battleships, fifteen armored cruisers, one second-class cruiser, one third-class cruiser, eight scouts, eighteen destroyers, eleven submarines.

It is proposed to begin during the financial year 1905–1906: One battleship, four armored cruisers, five ocean-going destroyers, one ocean-going destroyer of an experimental type, twelve coastal destroyers and eleven submarines.

Obsolete Ships—*Sale of.*—There was an immense concourse of buyers and spectators recently at the sale of obsolete war vessels at Chatham dockyard and the amount realized was over $690,000. Of the thirty vessels and other ships offered by auction, four were first-class cruisers, one a third-class battleship, three second-class cruisers and five third-class cruisers. The *Simoom* fetched $72,750; the *Warspite*, $90,750; *Northampton*, $79,000; *Arethusa*, $38,000; *Galatea*, $55,750; *Australia*, $54,500; *Severn*, $35,500; *Mersey*, $36,000; *Barracouta*, $20,000; *Exmouth*, $20,750; *Archer*, $24,250; *Cossack*, $24,000; *Mohawk*, $24,250; *Raccoon*, $20,750. The *Royal Adelaide*, hulk, fetched $17,500. "The Military and Naval Record" of April 6, says of this sale: "In connection with the sale of condemned warships, it is interesting to be reminded of the gigantic expenditure upon the battleship *Sultan* since she sank in the Comino Channel. This ship was completed in 1871 at a cost of $2,532,275, and she was considered a rather costly vessel for that period. Raising and repairing the ship cost $1,024,565, and it was considered by many naval officers at the time that the expense was not justified. Since then the money spent upon the *Sultan* has amounted to $662,850, so that her total cost to the nation has been nearly a million sterling. If memory serves us rightly, even more than a million was spent upon the *Inflexible*, the building of which was much delayed. The sale this week of

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unserviceable ships is likely to revive attacks upon the government, based upon expenditure incurred in the repair of ships now admitted to be either worn out or virtually obsolete. The government must submit to these strictures with becoming meekness, for it is an unquestionable fact that much money has thus been wasted."—"Army and Navy Register."

The vessels would undoubtedly have brought a better price but for the fact that, by the terms of the sale, they are to be broken up within a year.—(EDITOR.)

FRANCE.

Added to the Effective List.—The following vessels were added during 1904:

First-Class Armored Cruisers.—*Amiral Aube*, *Gloire* and *Condé*, all of 10,000 tons, 20,000 I.H.P., and 21 knots speed: *Desaix*, *Kleber*, of 7,735 tons, 17,000 I.H.P. and 21 knots speed.

Torpedo-Boat Destroyers.—*Arc*, *Baliste*, *Bélier*, *Dard*, *Francisque*, *Mousqueton*, *Sabre*, *Sarbacane*, all of 303 tons, with a speed of from 28 to 30 knots. These vessels are the last of their type to be completed. The new destroyers of the *Claymore* class are about 50 tons larger, and are to have a speed of 30 knots.

Eleven torpedo boats, up to No. 292, and eight submarines of the 68-ton class.

The principal ships completing or under trial are:

First-Class Battleships.—*République*, launched at Brest, 4th September, 1902; *Patrie*, launched at La Seyne, 17th December, 1903; *Démocratie*, launched at Brest 1st May, 1904; *Justice*, launched at La Seyne, 27th October, 1904; all of 14,865 tons, 18,000 I.H.P. and 18 knots speed.

First-Class Armored Cruisers.—*Léon Gambetta*, launched at Brest 26th October, 1901 (has commenced her trials); *Jules Ferry*, launched at Cherbourg, 28th August, 1903 (to be ready for her trials by the middle of present year); *Victor Hugo*, launched at Lorient 30th March, 1904; all of 12,550 tons, 27,500 I.H.P. and 22 knots speed; *Dupetit Thouars*, launched

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at Toulon, 5th July, 1901, of 9,516 tons, 19,600 I.H.P. and 21 knots speed (has now recommenced her trials, which were interrupted last March, owing to her having strained her hull).

The following large vessels are building:

First-Class Battleships.—Liberté, at St. Nazaire; *Vérité*, at Bordeaux; both of 14,865 tons, 18,000 I.H.P. and 18 knots speed.

First-Class Armored Cruisers.—Jules Michelet, at Lorient, of 12,550 tons, 27,500 I.H.P. and 22 knots speed; *Ernest Renan*, at Penhoet, St. Nazaire, of 13,562 tons, 40,000 I.H.P. and 23 knots; *Edgard Quinet*, at Brest, just laid down, of 14,300 tons, 40,000 I.H.P. and 24 knots speed.—"Journal of the Royal United Service Institution."

Léon-Gambetta—*Trials of.*—The repairs of the new first-class armored cruiser *Léon Gambetta*, rendered necessary by her mishap last month, have been rapidly carried out, and she has now at last successfully completed her trials, which were interrupted more than a year ago by the injuries the ship sustained through striking a pinnacle rock when steaming at high speed. The first trial was a preliminary one, and lasted eleven hours; only the central engine and four boilers were used, the results being satisfactory. The second preliminary trial was at high speed, the engines having to develop 25,000 I.H.P., or nine-tenths of her full power; this trial was for two hours, and the results were very satisfactory, the engines developing 26,200 I.H.P., 1,200 in excess of contract, the mean speed being 22.1 knots. During the 24-hours' coal-consumption trial the engines developed 16,911 I.H.P., giving a speed of 20.4 knots, with a coal-consumption of 761 gr. (1.52 pounds) per H.P. per hour. At the three-hours' full-speed trial the engines developed a maximum of 30,500 I.H.P., or 3,000 H.P., over the contract, and a mean of 29,008 H.P., giving a maximum speed of 24 knots—two knots over the contract—and a mean of 23.1, with a coal consumption of 176 kg. (387.9 pounds) per square meter of grate surface; the boilers of the large-tube Niclausse type worked most satisfactorily the whole time. The *Léon Gambetta* has thus proved

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herself the fastest cruiser as yet built for the French Navy, and she is the first large cruiser of the 12,600 type of the 1900 program, brought in by M. de Lanessan, to be completed. —"Journal of the Royal United Service Institution."

Dupetit-Thouars—*Trial of.*—The *Dupetit-Thouars*, a new French armored cruiser of 9,600 tons, 19,600 indicated horsepower, built at the Toulon Naval Yard, has recently completed her steam trials. Her machinery, supplied by Messrs. Schneider & Co. of Creusot, consists of three independent, vertical, three-cylinder, triple-expansion engines of equal power, each working a propeller. They are supplied with steam from twenty-eight Belleville boilers. These have a total of 113 square meters (1,216 square feet) grate area; a total heating surface of 3,580.04 square meters (38,540 square feet), and are subjected to a test pressure of 20 kilograms (294 pounds) per square inch.

The four series of trials made were the following:

1. *Coal Consumption on Trial at 1,800 Horsepower.*—According to the terms of the contract, this trial was to extend over six hours, the coal consumption to be from 650 to 700 grammes (1.4 pounds to 1.5 pounds) per horsepower hour.

The actual consumption during the trial amounted to but 519 grammes (1.1 pounds) per horsepower hour, thus showing a very satisfactory saving, the result of which will be to increase in a marked degree the radius of the action of the ship.

2. *Coal-Consumption Trial at 14,000 Horsepower.*—This was carried out on January 6, and lasted six hours. The consumption contracted for was 750 to 800 grammes (1.6 to 1.7 pounds) per horsepower hour. The actual consumption only amounted to 559 grammes (1.2 pounds) per horsepower hour.

3. *Maximum Power Trial.*—This was delayed for about one month, owing to slight damage sustained by the hull, the ship, under the action of a high wind, having come foul of the dock wing walls. The portside false keel was also torn by coming in contact with a mooring buoy. These damages were made good, and the trials were resumed on February 14.

The conditions to be fulfilled on this occasion were the following: Duration, three hours; power to be obtained, 19,600

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horsepower; combustion per square meter of grate area to be 160 kilograms (32.8 pounds per square foot) per hour.

The results obtained largely exceeded all expectations. The power developed reached 22,000 horsepower, while the rate of combustion did not exceed 140 kilogrammes per square meter of grate area (28.6 pounds per square foot) per hour.

The runs made on the new measured mile at the Hyères island in the Mediterranean showed an average speed of 22.05 knots, instead of the 21 knots contracted for.

4. *Trial at Normal Power (10,000 Horsepower).*—A recent naval order specifies that this trial, which lasts over twenty-four hours, has to take place not later than three days after the maximum power trial. The coal consumption contracted for was 750 to 800 grammes (1.6 pounds to 1.7 pounds) per horsepower hour. The actual results obtained were no less favorable than those of the preceding trials. The coal consumption amounted to but 590 grammes (1.3 pounds) per horsepower hour.

The admiral in charge of the trials expressed himself most highly satisfied with the performance of the ship. The results obtained and recorded above reflect the highest credit upon Messrs. Schneider & Co., and the Delaunay-Belleville Company.—"Engineering," London.

Liberté—*Launch of.*—On the 19th of April the latest of the French battleships was launched from the yards of the Ateliers et Chantiers de la Loire, near St. Nazaire. This ship is one of the *République* class of six ships, which will form a very powerful squadron in the French fleet, and will be distinctive from most of the other ships in that fleet, not only on account of the fact that they are the largest ships yet designed for the French Navy, but because of the fact that they are identical in design, whereas preceding ships have in almost every instance been of heterogeneous design, even though built at the same time and given approximately the same elements of military value. Being one of the latest of the *République* class, however, there are a number of improvements incorporated in the *Liberté* which are more in the nature of details

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than in the main features of the design. One alteration, however, which is of great importance, is the substitution of ten 7.7-inch guns for the sixteen 6.4-inch guns in the earlier ships. The *Liberté* embodies a thoroughly well-studied compromise of all the requirements imposed upon the designer of a modern battleship.

Navy	England.	United States.	France.	Germany.	Italy.	Russia.	Japan.
Ship	*King Edward VII.*	*Kansas.*	*Liberté.*	*Deutschland.*	*Vittorio Emanuele III.*	*Imperator Paul I.*	*Kashima.*
Displacement, tons	16,350	16,000	14,750	13,000	12,630	16,630	16,400
Length, L. W. L.	425 ft.	450 ft.	440 ft.	400 ft.	435 ft.	425 ft.	455 ft.
Beam, extreme	78 ft.	76 ft. 10 in	79 ft. 6 in.	72 ft.	73 ft. 6 in	78 ft.	78 ft. 2 in.
Draught, mean	26 ft. 9 in	24 ft. 6 in	27 ft. 6 in.	25 ft. 9 in.	25 ft. 10 in	27 ft.	26 ft. 4 in.
Horsepower	18,000	16,500	*18,000	*16,000	*21,000	18,000	21,500
Speed in knots	18.5	18	18	18	21	18	18.5
Armor, water line	9	9	11	9	10	11	9
for'd, aft	6— 3	4— 4	6— 6	7 and 4	6 and 4	6— 6	6.5— 6.5
citadel	8	7	10	5.5	8	?	6
casemate	7	7	6	6.75	None.	None.	6
deck	2	3	2— 3	3	4	2 75— 4	3
barbettes	12	10	11	10	10	12	9
turrets	12— 7	10— 6	12— 5	11	10— 6	12— 7	9— 5
Main battery	4-12 in. 4- 9 2 in. 10- 6 in.	4-12 in. 8- 8 in. 12- 7 in.	4-12 in 10- 7.7 in. ...	4-11 in. 14- 6.7 in ...	2-12 in. 12- 8 in. ...	4-12 in. 12- 8 in. ...	4-12 in. 4-10 in. 12- 6 in.
Secondary battery	14-12 lbr. 14- 3 lbr. 2 mach. ...	20- 3 in. 12- 3 lbr. 8- 1 lbr. 12 mach.	26- 3 lbr. 16- 1 lbr.	22- 3.4 in. 6- 3 lbr. 16 mach. ...	12- 3 in. 12- 3 lbr. 4 mach. ...	?	12-12 lbr. 12- 3 lbr. 6 mach. ...
Torpedo tubes	5-18 in.	4-21 in	5-18 in.	6-18 in.	4	4	5
Coal supply, tons	950-2,000	900-2,200	900-1,850	800-1,800	1,000-2,600	?	1,200-2,000
Crew	800	800	750	700	700	900	980

* Three screws.

In order to institute a comparison between the principal data of this ship and those of the latest battleships of other powers there is appended a table giving the principal characteristics of one battleship from each of the seven leading naval powers of the world, from which it is seen that the *Liberté* makes a very commendable showing in comparison with most of the others. If we compare her with the *Deutschland* of the German Navy, or the *Kansas* of the United States Navy, it is seen that her powerful guns could inflict serious damage upon the relatively thin water-line belts of these hypothetical enemies, while her own 11-inch armor belt would render her almost invulnerable to shells from the other ships. The same

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holds true when we compare her with the English *King Edward VII* and the Japanese *Kashima*, except for the fact that the two latter ships have one knot greater speed, and could therefore engage or not at will, or could choose the distance best suited to themselves in case an engagement was found desirable.—"Marine Engineering."

Aigrette and "Z"—*Comparative Trials.*—The "Moniteur de la Flotte" gives interesting particulars of the comparative trials of the submersible *Aigrette* and the submarine *Z*. The object was to take account of technical considerations touching surface navigation, habitability and, in particular, the method of plunging, and the time occupied in the operation. The *Aigrette* proved her absolute superiority when navigating on the surface, for she rose easily to the waves, and permitted her crew to enjoy a considerable degree of comfort. The *Z*, on the contrary, made heavy by her bridge, did not do so well. She has four torpedo tubes forward, which, when they are charged with torpedoes, add some two tons to the weight, this making the boat dip her nose in a very uncomfortable fashion, and causing the waves to wash the upper part from stem to stern. In plunging, the *Aigrette* did well, but neither boat had much advantage over the other. The conclusion arrived at was that the submersible alone possesses the right qualities, though the commission thinks that no sufficient reason for the submarine being abandoned. The submersible is the offensive boat, while the submarine is for coast defense. The experiments proved that the *Z*, which is of the Lutin type enlarged, both designed by the same constructor, has not gained superior qualities by the increase in dimensions, while the *Aigrette*, which is an improved Narval, possesses considerable improvements owing to her larger size. The committee considers that the submersible of the future should have a displacement of 400 tons, while that of the submarine should not exceed 100 tons.—"Army and Navy Register."

New Submarines.—The "Yacht's" Cherbourg correspondent states that orders have been received to put in hand two submarines to be known as *Q59* and *Q60*. They are large boats

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of the *Emeraude* type modified, and are to displace 425 tons, with a length of about 180 feet. The engines are to be of 1,200 horsepower, and the speed will be 12 knots. These boats, although spoken of as submarines, are in reality submersibles, and will, it is hoped, be capable of acting at long distance from the coasts. Sixteen submersible boats are to be apportioned for construction between Cherbourg, Rochefort and Toulon, and at the latter port is to be built a very small submarine known as *Q61*, which is to displace no more than about 21 tons. This will be an experimental boat of a class capable of being carried on board battleships or large cruisers. The correspondent says that success is much desired, but that no one is blind to the difficulties which are presented in the construction and employment of a submarine capable of being launched at sea. The operation might be conducted without much difficulty in sheltered waters, but in rough seas would probably be impossible. In any case considerable interest attends the construction of the small boat.—"Army and Navy Register."

Destroyers M40 to M43.—The new destroyers, *M40* to *M43*, are to have a displacement of 336 tons, with a length of 190.29 feet, a beam of 19 feet 9 inches, and a draught of 9 feet 6 inches; the engines are to develop 6,800 I.H.P., to give a speed of 28 knots. The coal supply will be 30 tons, giving a radius of action of 2,300 miles. The armament will consist of one 6-pounder Q.F. and six 3-pounder Q.F. guns, with two torpedo tubes for 17.7-inch torpedoes.

Sully—*Stranding of.*—The first-class armored cruiser *Sully*, a vessel of 10,000 tons displacement and speed of 21 knots, struck on a rock in Along Bay, in the Gulf of Tonquin, on February 6, while going out for target practice.

Efforts have been made to save her, but so far these have been unsuccessful, and it is now thought that she will be a total loss.

GERMANY.

Navy Estimates.—The total of the German Navy estimates for 1905 is M. 22,900,200. This includes a vote for construction which is subdivided to allow for the completion of the battle-

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ships *Hessen* and *Preussen*, the armored cruiser *York* and the small cruisers *Munchen* and *Lubeck*, and for the commencement of the two battleships *Q* and *R*, of the *Deutschland* class, the armored cruiser *D*, the small cruisers *O*, *Ersatz Wacht* and *Ersatz Blitz*, and two gunboats. There is a charge of a million and a half marks for submarines to be purchased. The total number of men voted was 36,622.

New Battleships.—The two new battleships *Hessen* and *Preussen* of the *Braunschweig* class are ready to begin their trials, and will probably be commissioned in the autumn of this year, and the *Brandenburg*, of an old type, has been reconstructed and entered into service again.

The small cruiser *Munchen* on her trials has made a speed of 22 knots with 10,580 H.P.

Several more vessels are expected to take the water from the German yards this year, although a large number were got off the stocks in 1904. The principal vessels will be the battleships *O* (building at Stettin) and *P* (building at Wilhelmshaven), the armored cruiser *C* (building at Bremen), and three protected cruisers, the *N*, *Ersatz Alexandrine* and *Ersatz Meteor*. These last-named vessels are building at the Weser Yard, at Kiel, and at Dantzic, respectively.—"Page's Weekly."

Ruebeck.—The German Admiralty has accepted the new turbine-propelled cruiser *Ruebeck*. The contract for this vessel called for 22 knots, but the *Ruebeck* made 23 knots an hour on her trial trip of 180 miles from Kiel to Swinemuende.

Cruisers.—The displacement of a cruiser, at present officially known as *C*, which was laid down in the course of last summer, will be 2,000 tons more than that of the *York*. The engines will work up to 26,000 horsepower, and the ship is to attain a maximum speed of 22½ knots. Another cruiser, officially known as *D*, is to be very similar to *C*. The turbine torpedo boat *No. 125* has made a trial trip of forty-two hours. The average speed attained was 28½ knots.

Three small cruisers, replacing the *Meteor*, the *Alexandrina* and the *U*, which, in the course of the present financial year,

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have been laid down in the Imperial shipyards at Kiel and Dantzic and the yards of the Weser Company, are of the same type as the small cruisers *Bremen*, *München* and *Hamburg*, already completed. The *U* is to be finished by the spring of 1906, and the *Meteor* and the *Alexandrina* by the spring of 1907. Three other small cruisers—the *O*, the *Wachl* and the *Blitz*—have been projected. They are of the *Meteor* type and are to be 366 feet long by 44½ feet beam. They are to have a displacement of 3,400 tons each.—"Engineering," London.

ITALY.

Regina Margherita—*Trial.*—The trials of the *Regina Margherita* have been completed with fairly satisfactory results. On her first trial, at 14,000 H.P., she attained a speed of 19.3 knots, and later, with 20,600 H.P., the speed was 20.2 knots. It will be noted that for an increase in power of nearly 50 per cent. the increase in speed was slightly less than one knot, and this is a matter the Italian authorities might well give attention to. With some of our County cruisers a rather similar experience occurred, and it was not until the propellers had received attention that the vessels were able to give such excellent performances as they are now capable of. The *Regina Margherita's* boilers are of the Niclausse type.—"Page's Weekly.

New Ships.—Particulars of the design of the new armored cruiser *A*, which is to be built at Castellamare, have been published and show that this vessel will be in length 425¾ feet; beam, 68¼ feet; draught, 23¼ feet. The maximum thickness of the belt will be 8 inches, while the guns carried will be four 10-inch in pairs in turrets fore and aft, eight 8-inch in pairs in turrets on the broadside, sixteen 3-inch and eight machine guns. The engines will develop 18,000 horse-power, giving a speed of 22.5 knots, and the coal carried will be from 700 to 1,500 tons. Her displacement will be 10,000 tons, and the cost is estimated at £880,000.

A sister cruiser will be built at the same time and at the same yard. The first cruiser will probably be named the *San Georgio*,

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but the second is known at present as *B* only. The names of the torpedo boats building for the Italian Navy are as follows: At Schichau, the *Sirio*, *Sagittario*, *Spica*, *Scorpione*, *Serpente* and *Saffo*. At Odero, the *Orione*, *Orsa*, *Olimpia*, *Orfeo*, *Alcione*, *Ardea*, *Albatross*, *Aibone*, *Astore* and *Arpia*. At Pattison's yard, *Pegaso*, *Perseo*, *Procione*, *Pallade*, *Cigno*, *Cassiopea*, *Calilope*, *Clio*, *Centauro* and *Canopo*. At Spezia, the *Gabbiano*.—"Page's Weekly."

JAPAN.

Kashima—*Launch of*.—Sir W. G. Armstrong, Whitworth & Co., launched from their Elswick works on March 22d the third battleship and the tenth war vessel built there for the Imperial Japanese Navy. This latest vessel is named the *Kashima*, and is one of two ordered at the beginning of last year by the Japanese Government; the other is being built by Messrs. Vickers Sons & Maxim, Limited, at the Naval Construction Works, Barrow-in-Furness. The *Kashima* is the most powerfully-armed vessel yet constructed in this country. She has four 12-inch guns, twin-mounted in barbettes, the upper part being of 9-inch plating, and the lower part, where protection is augmented by the ordinary citadel armor, of 5-inch armor. These 12-inch guns, which have hydraulically-operated mountings, are 46.7 calibers long, and weigh 59 tons. Firing 850 pound projectiles with modified Cordite explosive compound, it is anticipated that the penetrating power developed will be such as to cope at 3,000 yards range with any armor yet fitted to a battleship, and that the rapidity of fire will exceed two rounds per minute from each gun. There are also four 10-inch 34-ton guns of 46.76 calibers, placed singly in barbettes, built up of 6-inch plates, located at the four corners of the citadel. There are twelve 6-inch 8½-ton guns of 47 calibers. Five are mounted on each broadside inside the 6-inch citadel armor, and are separated from each other by 2-inch screen armor. The two remaining 6-inch guns are on the upper deck within 4-inch screen armor. There are also twelve 12-pounder guns, three 3-pounder guns, and six maxim guns.

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In the design of the arrangements for serving the guns the lessons of the recent war have been utilized. There are five submerged torpedo tubes, firing 18-inch Whitehead torpedoes; two are forward and two aft, firing on the broadside, with one firing right astern. As regards the disposition of armor, the principle adopted is generally similar to that in the *Mikasa*, the most recently completed vessel for the Japanese Navy. The water-line belt is 9 inches in thickness for more than half the length of the ship, tapering to about 4 inches at the ram. This belt extends 5 feet below the water-line and 2 feet 6 inches above it. Surmounting it is a belt of armor, extending from the aft 12-inch barbette to the stem, and 6 inches in thickness. Immediately above this is the 6-inch citadel armor, reaching to the upper deck and inclosing also the two 12-inch barbettes. It is within this that the 6-inch guns are placed, while above this, again, is 4-inch screen armor for the two upper-deck 6-inch guns. The conning tower is of 9-inch armor, and there is an additional observation tower of 5-inch armor, while two more officers' shelters of 3-inch armor are provided on the boat deck amidships. In this way the fighting officers of the ship will have very advantageous positions, protected against the fire of the enemy's small guns.

The protected deck is 2 inches on the flat portions amidships and 3 inches on the sloping sides, where it joins the bottom of the main armor 5 feet below the load line. At the extremities, where the side armor is reduced in thickness, this deck is made 2½ inches thick over all. Further protection is given to the upper structure of the ship by thick protective plating on the top of the screen armor at the level of the boat deck. The length of the *Kashima* is 455 feet; the breadth, 78 feet 2 inches, and the moulded depth 43 feet 6 inches. With a normal draught of 26 feet 7½ inches the displacement will be 16,400 tons; the coal supply then will be 750 tons, but capacity is provided for 2,150 tons. Triple-expansion machinery and twenty Niclausse boilers are being constructed by Messrs. Humphrys, Tennant & Co., London. The designed

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power is 16,500 horsepower, which, it is anticipated, will give the vessel a speed of 18½ knots.—"Engineering."

Kiji—*Launch of.*—The torpedo-boat destroyer *Kiji*, 200 tons, was launched at Kure, Japan, April 19.

MISCELLANEOUS.

Yarrow-Napier Motor Boat.—On March 4th an exhibition trial was made on the Thames, at Greenwich, of a little boat which is said—no doubt accurately—to have a speed considerably in excess of that of any other vessel of equal length. The next fastest boat in the world, of this class, is the American *Challenger*, the mean speed of which is stated by the United States press to be 25.01 knots. Next to this is the *Tréfle-à-Quatre*, a French boat, which has accomplished a speed of 22.7 knots. The Yarrow-Napier motor boat is 40 feet long, and is fitted with two four-cylinder petrol engines, driving twin-screws, and giving collectively about 120 horsepower. The boat has been constructed by Messrs. Yarrow & Co., and the engines by Messrs. Napier & Co. She is, we believe, intended to be used by Mr. S. F. Edge for racing purposes, as she is to be dispatched to Nice, where she will doubtless take part in the forthcoming competitions in the Mediterranean.

A trial of this little vessel had been previously made on the measured mile in Long Reach, when a speed of practically 26 knots was reached. The run against tide was made in 2 minutes 25 seconds, and the mile with tide was covered in 2 minutes 12.6 seconds. The mean speed was therefore 25.988 knots, or just on 30 statute miles an hour. The result is more remarkable as the day was by no means favorable to a good performance, for there was, according to the meteorological reports, half a gale of wind blowing at the time, and even in the inclosed waters of Long Reach this would make a considerable difference to the speed result for so small a vessel. In reference to the fraction of a second recorded, it may be stated that the time was taken by the official timekeeper of the Automobile Club, Mr. Swindley, who, of course, is accustomed to deal with decimal parts of seconds, and was

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confirmed independently by Mr. Marriner, of Messrs. Yarrow's staff, who is also an expert in taking time records.

The hull is constructed of what is described as a special steel, which is probably of the description used by Messrs. Yarrow in their torpedo craft and has had so much effect in lightening the structure. The framing is on the longitudinal principle in order to secure the maximum strength fore and aft. Naturally, the scantling of the hull in a petrol motor boat may be much lighter than in a steam-driven vessel, as there is no boiler to add to the displacement, and the weight of fuel is considerably less—about one-third that of coal.

As already stated, the length of the boat is 40 feet, the width is 5 feet, and the draught of water of the hull is 8 inches. The total weight of the boat in running trim is 3 tons and 4 hundredweight, of which the machinery weighs approximately one-half and the hull the other half. The engines have each four cylinders, or eight in all, the dimensions being 6½ inches for the diameter and 6 inches for the stroke. Each set of engines drives a three-blade propeller.

The form of hull is unusual, the shape having been arrived at by means of a long series of experiments made by Messrs. Yarrow with full-sized models towed by a torpedo vessel. The bottom for the greater part of the length is quite flat, the bilge being only just rounded off. The bow is by no means sharp, and there is no gripe forward, the forefoot being quite blunt; but this is of little consequence once high speed is reached, as the bow lifts right out of the water for a considerable distance, the vessel running entirely on the flat part of the bottom. This is a condition purposely secured, the idea being to make the boat rise bodily in the water when traveling at high speed. In the early days of torpedo-boat construction it was noticed that the displacement was reduced somewhat when the highest speed was reached; but, naturally, with the greater weight of steam machinery this effect was hardly a practical factor, excepting at a great expenditure of power. Many of our readers also will doubtless call to mind the suggestions made about thirty years ago by a clergyman, the Rev. C. Ramus. His con-

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tention was, if we remember rightly, that the way to get the greatest speed possible from a vessel would be to make her of such a shape that she would offer excessive resistance to passage through the water, and therefore she would rise bodily out of the water when great power was exerted by the engines, so that the friction and wave-making features would be reduced. Mr. Ramus was a good deal laughed at at the time, but in view of Mr. Yarrow's experiments he appears, like so many other inventors, to have only been thinking thirty or forty years before his time; but although his principles may have been sound within limits, he was quite visionary in attempting, as we believe he did, to force them on the Admiralty for the construction of warships. It would have been pleasant, however, if the reverend inventor could have lived to see the really wonderful performance of the little boat in Greenwich Reach on Saturday last.

Amongst the models tried by Messrs. Yarrow—and they were, it must be remembered, the full size of the actual boat—the one which gave the least resistance at speeds aimed at was flat from end to end; in fact, no "shape," as the boat builder understands the term. The hull appeared to glide on the surface of the water, and the recording instruments indicated that such was approximately the case. This points to the fact, as Mr. Yarrow has stated, that the power exerted by the engines were better devoted to supporting the boat on the surface rather than by putting it through the water. With a flat-bottom boat the speed may be such that the water, which would be displaced were the boat at rest or traveling slowly, has not time to move downwards out of the way before the boat has passed, and as no two things can be in the same place at the same time, the boat has to rise bodily. That this will occur to some extent, even with ordinary ship-shape vessels, when the speed is excessive, is well known, and, with a flat-bottom craft, where the movement of the water must be at first vertical, the action will evidently be intensified. This was well illustrated by Mr. Yarrow by means of the analogy of a flat stone being thrown on the water, making what children

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style "a duck and a drake." Evidently the stone would travel further and faster than if the same energy were expended in projecting it partially submerged. Considering the speed at which the boat traveled—or, perhaps, for reasons stated, it was on account of the speed—the waves thrown off appeared very small, although those that were thrown off were steeper than usual. It was difficult to judge, on account of the disturbed state of the river, and it would be interesting to see the boat tried on a perfectly calm day. If runs could be made at the best speed in deep water, and also in water so shallow as only to give a sufficient margin of draught just to float the vessel, some interesting results might be obtained. Possibly a safe stretch, with a soft bottom, might be found in the estuary of the Thames or Medway.

In this boat, as stated, a considerable length of the forward part is clear of the water when she is traveling at her best speed. Unfortunately, there was too much popple on Saturday to estimate the distance, but evidently it was something considerable. It would naturally add to the speed in quite smooth water if the forward out-of-water length were sawn clean off, as the weight would be reduced; but that would give a very bad bow for waves to strike against, and it is possible the initial resistance might be too great. It may be suggested, however, that a cone-shaped bow might be of advantage, as it would be lighter, and oppose less projected surface to the blow of the waves than even the orthodox section.

The performance of this boat is exceedingly interesting, as it gives information regarding what takes place at extremely high speed in a vessel propelled by its own screw or screws. It is not difficult to arrive at conclusions in this field of research by means of tank experiments, leaving the disturbing action of the screw out of account; but that does not represent the actual condition of affairs in a ship at sea, when the propeller must have considerable influence on the stream lines. To arrive at some conclusions in this respect by tank experiments, a model has had a screw placed in the usual position

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and driven by an outside means, but to have recourse to such a plan obviously is attended with considerable difficulty.

Mr. Yarrow has attacked this problem with his usual thoroughness and appreciation of the practical needs of the situation, and has added one more to the long list of benefits he has conferred on the science of marine propulsion. He states that had a boat of similar dimensions to that exhibited on Saturday been required to be driven by steam, his firm would not have been prepared to promise more than 16 knots as a maximum; so it would appear that the substitution of the petrol engine for a steam engine and boiler has resulted in a gain of no less than 10 knots.

Mr. Yarrow estimates, on the basis of the performance of the present boat, that a destroyer of 220 feet with petrol engines would attain a speed of 45 knots. The destroyers of this length built by Messrs. Yarrow for the Japanese Government steamed at 31 to 32 knots. As Mr. Yarrow remarked, all that remains to be done in order to reach the 45 knots is to perfect the internal-combustion engine, so as to enable larger sizes to be used successfully.

Such advances are without precedent in the history of marine propulsion; and they hold out such rewards that it can hardly be doubted every effort will be made to adopt the internal-combustion engine for still larger vessels. The difficulties in the way are numerous, the most prominent one at present being the high price of the fuel used; but beyond this there are many physical and mechanical troubles to be overcome before internal-combustion engines of large size can be used at sea.

The saving in weight of fuel is, of course, considerable, it being placed by Mr. Yarrow at two-thirds that of coal for ordinary steam machinery of the same power. The addition this would give to the radius of action is a military advantage, the value of which is apparent for torpedo craft; whilst it would render destroyers far more capable of accompanying a fleet, as, for a given distance to be covered, they need not be weighted down to the same extent, and would be more sea-

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worthy. In the boat tried on Saturday there is a petrol tank of a capacity of 100 gallons—a quantity sufficient to take the boat 130 miles at full speed, or double that distance at three-quarter speed.—" Engineering," London.

Roosevelt.—The vessel which Civil Engineer R. E. Peary, U. S. Navy, will use in his attempt to reach the North Pole has been launched, as has already been announced in these columns, and is well on the way toward completion. The designs were made by William E. Winant, of the Bureau of Construction and Repair in the Navy Department. All the details of the vessel take into consideration that peculiar work upon which he will be engaged. The dimensions of the *Roosevelt*, as the vessel has been named, are as follows: Length on load-water line (between perpendiculars), 161 feet; length on deck (about), 172 feet; length over all, from tip to tip of rail (about), 182 feet; breadth, molded (to outside of plank at deck beam), 33 feet; breadth (to outside of plank at L. W. L.), 32 feet; breadth, extreme, over guard, 34 feet 2 inches; depth, molded (from top of main-deck beam at side to bottom of keel at midship frame), 20 feet; draught, forward, 15 feet 3 inches; draught, aft, 16 feet 9 inches; mean draught, amidships, 16 feet. Her displacement, when ready for sea, with all coal and stores aboard, will be about 1,500 tons.

Probably in no part of the world can be found a model or form of hull similar to this vessel, whose mission is to drive into, break down and force away the ice fields in front, with a stern so shaped that the overhanging portions will more or less protect the screw when the heavy ice floes come together against the vessel's quarters. There is no "tumble-home" of the top sides such as may be found in former Arctic vessels, but instead the top sides "flare out" up to the rail, which admits of working a heavy "guard strake."

The purpose of this guard strake, apart from being a buffer, is to help to lift the vessel out of the water as the ice crushes about her. As can be seen, the ice will press against her sides, and then, as they offer first resistance, it will rise and catch under the counter or overhang of this guard and bodily raise

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the vessel. In case the ship, on the other hand, has been frozen in a thick pack of ice, and it is desirable to free her, hydraulic jacks are set upon the ice and brought to bear upon the under side of the guard, and these, in turn, will raise the craft, and as she is permitted to settle back her own weight and her form, acting as a big wedge, will tend to break away clear. This is a peculiarly novel arrangement.

Reference to the midship section and the lines will show the combination in form of the old-time "frigate floor" and the modern ferryboat bilge, which, together, gives a body that will admit of the vessel laying over on her side without damage, and also gives what is known in ship parlance as "freezing section," from lateral compression of ice from without.

The most important feature of this vessel's form is the bow and fore-foot. This is designed so that when driving into thick ice the vessel will ride upon the ice and break it down and at the same time keep the broken ice from piling so high as to be an impediment to the next ramming. The extreme flare of the bows from the water line to the forecastle deck is for this purpose also.

The *Roosevelt* is braced, girded and reinforced as never was craft before; she will be able to withstand enormous pressures from surrounding fields of ice. This "squeeze" of the ice will tend to lift rather than crush her between the grasp of the gathering pack. The extreme "ferryboat" form of midship section will raise her up out of the ice the harder the squeezing becomes.

The vessel may be technically described as a "three-masted, fore-and-aft, schooner-rigged steam vessel, with auxiliary sail power," larger than Nelson's famous *Fram*, but about the size of the Antarctic ship *Discovery*. There the similarity ceases.

Continuity of strength, fore and aft, laterally and vertically, only have been considered in view of the work to be accomplished, as may be noted from the midship section and inboard profile. When it is considered that the bow framing extends ten feet aft of the stem and is of solid dead wood, etc., some idea of the terrible pounding this vessel will stand can

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be appreciated. This solid framing is from the natural white oak crooks and is bolted from all directions. In addition to this heavy framing a three-quarter-inch vertical apron plate is worked inside of all to prevent the forward hood ends of the double planking from "starting" during the continual ramming into the ice. The archives of shipbuilding fail to show a vessel so strongly framed as the *Roosevelt.*

Compared with the United States Government revenue cutters *Bear* and *Alert* and with Arctic whalers of recent years, the *Roosevelt* is a much stronger vessel throughout. She is double-framed and double-planked with two courses of 5-inch oak plank and ceiled on the inside of the frame with 3-inch yellow-pine ceiling. In addition to this heavy framing, big 14 x 22 inches "'tween"-deck beams are worked in every frame at the water line to prevent lateral crushing of the vessel.

Between the main deck and the "'tween"-deck beams heavy athwartship diagonal braces are worked to prevent the vessel being crushed from the bilges upward.

The longitudinal strength is maintained in each side by very heavy oak and yellow-pine keelsons, clamps, stringers, deck plank and inside and outside planking, all of which is copper bolted in all directions.

To take up any vertical working of the vessel that may occur from months and months of forging into and breaking down the ice, heavy wrought-iron tie rods are fitted from the lower side of the main keel to the top of the deck beams. These tie rods are run through wrought-iron pipes (which act as beam stanchions), and they may be "set up" at will on the main deck.

To take the rub and grind of the floating ice, the bow, sides along the water line and the stern are protected by steel plating. At the stem this plating, an inch thick, reaches from the keel all the way up to about 6 feet above the load-water line and extends aft from the stem for quite 12 feet. This is for protection of the bow when ramming. The water-line plating, which is $\frac{3}{8}$ of an inch thick, is a continuous belt 5

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feet wide, 1 foot only above water, the 4 others being below, where they will be more serviceable. The fact that this steel plating should have to reinforce the unusually heavy double course of planking is significant of the tremendous wear and tear to which a craft is subjected in the polar seas.—"Army and Navy Register."

Ferry Launch for Navy Yard, Portsmouth, N. H.—Plans have been made for a ferry launch to be used at the Portsmouth, N. H., Navy Yard. The dimensions of this boat will be—

Length over all, feet	80
on load-water line, feet and inches	70– 9
Beam, outside of planking amidship, feet	20
Draught, feet and inches	5–11½
Displacement, tons	80

A 300-horsepower gasoline engine will be installed to propel this boat. The engine will have six cylinders of 12 inches diameter by 14 inches stroke and will weigh 8,800 pounds. The total weight of machinery, including tanks, will be about 12,025 pounds. The gasoline tanks will hold a supply of 500 gallons. A small hot-water heater with radiators in cabins and pilot house will furnish heat during the winter months.

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MERCHANT SHIPS.

Amerika—*Launch of.*—On Thursday, April 20th, Messrs. Harland & Wolff, Belfast, launched the large steel twin-screw passenger steamer *Amerika* for the Hamburg-American Line. The new steamer has a gross tonnage of about 22,800 tons, and, when completed, will be the largest vessel afloat outside the British mercantile marine. She will be fitted for a large number of first, second, intermediate and third-class passengers. Several new features have been introduced, the chief of which are the special restaurant and a fine recreation room containing a gymnasium and other facilities for athletic exercise. The machinery is of the builders' quadruple-expansion balanced type, reducing vibration to a minimum.

La Provence.—On the 21st of March the latest flyer of the French line was successfully launched near St. Nazaire, in the presence of the Ministers of Commerce and Public Works and the First Secretary of Marine. The *Provence* is of a considerably larger size than the ships of the French line at present in service, and, indeed, is the largest ship of the French mercantile marine, and the largest ship ever built in France.

She has a length of 626 feet, with beam of 69 feet, and displaces, at a draught of 26 feet, 19,200 tons. She is fitted with two engines, operating twin screws, and developing a total of 30,000 indicated horsepower, with which it is expected that a speed of 23 knots will be maintained. The draft has been limited because of the fact that the harbor of Havre, which is the eastern terminus of the trip of the French liners, is relatively shallow, and will not admit of the draught given the English ships entering the port of New York. The ship was commenced in December, 1903, and is expected to be ready for service early in 1906. It was proposed, when the plans were being considered, to fit her with turbine engines, but the Companie Generale Transatlantique had scarcely the courage to initiate this system in a steamer of such importance. Her

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crew numbers 439 all told, and she has provision for 1,900 passengers.—"Marine Engineering."

Dieppe.—The London, Brighton and South Coast Railway Company's new turbine steamer *Dieppe* was launched on Thursday, April 6th, by the Fairfield Shipbuilding and Engineering Company, Limited, Govan. The vessel is very similar in general dimensions to the same company's *Brighton* (see "Engineering," vol. lxxvi, page 325), which was launched by Messrs. William Denny & Bros., Dumbarton, in June, 1903; but on the later vessel a considerable number of improvements have been made. The beam has been increased, and the after portion of the ship, which is open in the *Brighton*, has been closed in, thus giving greater comfort and protection to the second-class passengers and insuring still further the safety of the steamer. The *Dieppe* is 274 feet in length, 34 feet 8 inches in breadth and 14 feet 6 inches in depth. The first-class dining saloon, extending right across the ship, gives dining accommodation for 62 passengers, and there is sleeping accommodation for 36. The second-class sleeping saloons are on the lower deck, and accommodate nearly 200 passengers. The propelling machinery consists of three independent Parsons compound steam turbines and two condensers. Each of the three turbines drives a separate shaft, with one three-bladed propeller on each shaft. In order to cope with the exigencies of the service, the builders have designed a special starting, reversing and maneuvering gear, which is quick in its action, while the movements of all the turbines can be controlled from the starting platform by one engineer. The boilers are four in number, single-ended, of the ordinary multitubular marine type, and fitted with Howden's forced draft. They are constructed entirely of steel, and adapted for a working pressure of 150 pounds per square inch. The turbines and other machinery have been constructed by the Fairfield Company.—"Engineering," London.

James C. Wallace—*Launch of.*—While the *Wallace* is 8 feet shorter than the steamer *Augustus B. Wolvin*, the largest vessel on the lakes, the cubical capacity of her cargo hold is actually greater because it is four feet wider at the top and

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five feet wider at the bottom, while only being one foot less in length. The *Wallace* has thirty-two hatches as against thirty-three in the *Wolvin*. In all other respects the steamers are structurally duplicates. The *Wallace* is 552 feet over all, 532 feet keel, 56 feet beam and 31 feet deep. The hopper extends in one continuous length of 408 feet without bulkheads or divisions of any kind, and in width it measures at the top 47 feet, and at the bottom 29 feet. The space between the sides of the ship and the sides of the hopper is used for water ballast, so that the water ballast, as well as being in the usual double bottom underneath the hopper, extends up the sides to the height of the main-deck stringer. This construction was adopted to accommodate the use of automatic clam shells in unloading the cargo, and while it serves this purpose, it also serves to make a stronger vessel. Another feature of the construction is the fact of the ordinary hold stanchions being dispensed with, and in their place a system of girder arches is substituted to support the upper deck as well as the sides of the vessel. There are thirty-two cargo hatchways on the spar deck, each one measuring 33 feet by 9 feet in the clear, and spaced 13 feet apart centers, so that it will be possible for the clam shells to unload the vessel entirely without any hand shoveling. These hatchways are fitted with a patented system of steel-plate sliding-hatch covers known as the Brousseau device, operated by steam engines and shafting, so that no manual labor is required for the opening and closing of the hatch covers.

The spar deck is continuous and flush from stem to stern, and the only erections on it are the pilot house and texas forward, and the coamings around the engine and boiler openings and dining-room skylight aft. The water-ballast space is divided into numerous compartments by watertight athwartship bulkheads fitted at intervals of about 60 feet, and the total ballast capacity is about 8,000 tons of water. On the spar deck are six 8 by 10-inch single-drum engines, carrying each a steel-wire mooring line for the usual mooring and warping purposes. In fact, steam has been used wherever possible to displace manual labor. There is a steam windlass fitted

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for handling the two 8,000-pound anchors, and a steam capstan on the spar deck aft. She will be fully equipped with electric lights throughout, and has duplicate plants, so that there will always be one to fall back on in case of the other giving out. In the forward end of the vessel, immediately under the pilot house and texas, there are five staterooms for owners and their families, also accommodations for their private galley, dining room and social parlor; also two toilet rooms. They are entirely set apart from the crew's quarters and will be luxuriously furnished and fitted out. The ship's officers are accommodated on the deck immediately below the owner's quarters, and the engine-room officers are quartered alongside of the engine at the after end of the vessel, where is also located the kitchen, dining room and mess room for the crew.

The propelling machinery, as is usual in lake vessels, is located in the extreme after end of the boat. The main engine is of the quadruple four-crank type, with cylinders 18½ inches, 28½ inches, 43½ inches and 66 inches diameter, each having 42 inches stroke of pistons, and they are designed for an indicated horsepower of 2,000 when making 80 revolutions per minute. The cylinders are arranged from forward in the following order: High-pressure, first-intermediate, second-intermediate and low-pressure. The high-pressure is fitted with a liner secured in the usual way. The valves of the high, first and second intermediate are of the piston type, placed in front of their respective cylinders, and driven by "Joy" radial valve gear. The low-pressure cylinder is fitted with a double-ported slide valve on the after end, operated by double-bar Stephenson link motion. The bedplate is of the girder type, of box section, in two pieces, bolted together, and has five main bearings lined with white metal. There are four back and five front columns, the former acting as guides for the crossheads. The piston rods are of steel 5⅜ inches diameter, and the connecting rods of hammered iron 9 feet long between centers, with wedge adjustment at top and "T" ends with bolts at bottom. The crossheads are of cast steel, keyed to piston rods and fitted with steel crosshead pins, secured with double taper and nut. The slippers are of bronze of large

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surface and easily adjustable. The crank shaft which, with its four cranks is built without couplings, is 12½ inches diameter. The crank pins are each 12½ inches diameter by 13½ inches long. The thrust is of the "horseshoe" type with six go-ahead collars and five backing. The engine is fitted with a direct-acting steam, and a hand, worm reversing gear, also a steam turning gear at aft end of thrust shaft. The propeller is four-bladed cast steel, of sectional type 14 feet 6 inches diameter and 14 feet 6 inches pitch. The outboard bearing is 13 inches diameter by 4 feet 8 inches long, filled with lignum vitae.

The boilers, two in number, are of the Babcock & Wilcox type, and placed with a 12-foot space between them to fire athwartship. The working pressure is 250 pounds per square inch, and the steam is superheated, the superheater being placed on top of boilers proper. The boilers are fitted with the mechanical stokers built by the Duluth Stoker Co., the coal being fed into the hoppers, passing from thence on to traveling grates, the ashes being deposited at the back of boilers and discharged overboard by means of steam-driven elevators. The boilers are fitted with a system of induced draft, having two 7-foot 6-inch diameter fans driven by double 6 by 5-inch high-speed engines. To purify and heat the feed water before reaching the boilers, two Learmonth purifiers are fitted, also a large heater into which all the auxiliaries exhaust. There are the following auxiliaries in the engine room, the pumps, all being of the Blake type: One vertical cross-compound simplex air pump; two horizontal cross-compound simplex ballast pumps; one horizontal compound duplex feed pump; one horizontal duplex pony pump; one horizontal duplex bilge pump; one horizontal duplex cooler pump; two general electric generating sets of 15-kw. for electric lighting; one Globe steam steerer and one Sturtevant fan with attached engine for ventilating purposes.

The coal bunker is fitted in front of the boiler room and has capacity for about 350 tons of coal. In every respect the vessel has been fitted out and built to meet the most modern requirements of lake service.—"Extract from Marine Review."

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YACHTS.

New Turbine Yacht for the King.—The order for a new yacht for the King has been placed with Messrs. A. and J. Inglis, Pointhouse, Glasgow, who have a very high reputation for the design and construction of such craft. It will be the first Royal yacht constructed by a private firm within recent years. The Admiralty adopted the very wise course of giving the firms invited to submit designs a completely free hand, and the designs carried only a *nom-de-plume*. The committee which was subsequently appointed to adjudicate upon the design included the Commodore of the Royal Yachts, Sir A. Berkeley-Milne, the Director of Naval Construction, Mr. Philip Watts, and Admiral Sir John Fullerton, who was for long the Commodore of the Royal Yachts. They had no knowledge as to the origin of any of the designs. This committee, we understand, were unanimous in their preference for the Inglis design, and when their report was submitted to the King, he also, after careful examination of the general arrangements, acquiesced in the decision.

The new yacht is primarily intended for short cruises and for visits to harbors where the draught of water is limited, as in the case of Flushing, Nice and some other Continental ports. The size of the ship has therefore been limited, although a cruising speed of 17 knots is expected, with a go-astern speed of 15 knots. The vessel will be 285 feet long on the load-water line, 40 feet in breadth, and of 2,000 tons gross. The dining room, the Queen's drawing room and the smoke room will be on the upper deck aft; and it is interesting to note that the decorative work, furniture, &c., will, as far as possible, be made at Glasgow, showing that this city has advanced very considerably in artistic production. The propelling turbines will be of the Parsons type, and in accord-

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ance with the now almost universal arrangement there will be three shafts—the high-pressure turbine being in the center, and the low-pressure turbine for going ahead on the side shafts, and abaft of them the go-astern turbines. Cylindrical boilers are to be adopted for generating steam.—"Engineering."

Albion.—The triple-screw turbine yacht *Albion*, which has been a feature of such interest on the Tyne for some time past, sailed on Thursday night and arrived early on Saturday morning at Southampton. The yacht has been built by Messrs. Swan, Hunter & Wigham Richardson, Limited, for Sir George Newnes, Bart, M. P., and is from the design of Sir W. H. White, K. C. B. The machinery consists of turbines by the Parsons Marine Steam Turbine Company, driving three propellers. Before the vessel left the river she went through a long series of exhaustive trials, from which were gathered data which will prove of great value in the application of turbines to similar vessels.—"Page's Weekly."

On Thursday, April 20th, Messrs. Ramage & Ferguson, Limited, Leith, launched a twin-screw steam yacht of over 1,000 tons, Thames measurement, for Baron De Forest. The vessel is constructed on the double-deck principle, so popular with some yachtsmen nowadays, the shade deck being continuous right fore-and-aft, and the sides built up, except for a part aft of the forecastle and another part aft in way of the gangways. The accommodation is, perhaps, the largest ever attempted in a yacht of this tonnage. The dimensions are: 260 feet long over all, and 213 feet on the water line, 31 feet beam and 16 feet deep. She will be rigged as a pole-masted schooner. Immediately forward of the machinery space on the cabin deck is a large stateroom, etc., for the owner, with a stairway leading to the main deck, communicating with a private boudoir and a large dressing room. Just aft of these rooms on the main deck is a spacious dining room, and on the starboard side of the boiler casing there is a library and an anteroom, etc., through which the dining room is entered. On the port side of casings are servants' rooms, larder, etc., and in the

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casings themselves are the pantry and galley. Aft of the engine casing on the main deck is a large deck house, forming a drawing room, with a vestibule leading to the cabin deck. In the poop are three good staterooms, bathroom, etc., and on the cabin deck aft of the machinery space are large staterooms, with bathrooms, servants' rooms, etc. On the shade-deck forward is a large deck house, containing a sitting room, chart room and captain's room, with navigating bridge above. Aft on the shade deck there is also a deck house, forming another sitting room.

The yacht is not intended to be a speedy craft, but will cruise at a speed of 13 knots. She will be propelled by twin-screws driven by two sets of triple-expansion machinery, embodying all the modern improvements, and capable of indicating 1,400 horsepower. There is one large single-ended main boiler, having a working pressure of 180 pounds, and a Blake donkey boiler for work in port. She will have a coal-carrying capacity of about 230 tons, which should enable her to cross the Atlantic without recoaling. The funnel is provided with a patent spark catcher. There will be a complete electric-light installation, Hall's ice machine and all up-to-date conveniences. This vessel is the last designed by the late Mr. G. L. Watson.—"Engineering," London.

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OBITUARY.

ASSISTANT NAVAL CONSTRUCTOR JOSEPH E. McDONALD, U. S. NAVY.

Assistant Naval Constructor Joseph E. McDonald was born in Indianapolis, Indiana, on February 17, 1872. He was appointed a Naval Cadet on September 7, 1888, and was graduated from the U. S. Naval Academy in May, 1892.

While at the Academy Cadet McDonald proved himself to be possessed of unusual ability and naturally assumed a position of leadership amongst his companions. During his final year he held the highest cadet office in the battalion.

After graduation Cadet McDonald was ordered to the U. S. S. *Newark*, but was shortly afterwards sent to Paris, where he spent three years as a student of Naval Architecture.

On July 1, 1894, he was commissioned an Assistant Naval Constructor, and was first assigned to duty at the works of the Newport News Shipbuilding and Dry Dock Company, Newport News, Virginia. He afterwards served for several months on board the U. S. Ships *New York*, *Maine*, *Texas* and *Indiana*, and on June 3, 1897, returned to Newport News, where he reported for duty as the Assistant Superintending Naval Constructor. On October 26, 1898, he was transferred to Richmond, Virginia, in connection with the construction of several torpedo boats and destroyers, and in June of 1899 was ordered to the Navy Yard, Mare Island, California, where he remained until granted a sick leave of absence in January, 1902.

This leave of absence was extended from time to time in the hope that his health might finally be restored. In November of 1904 he was ordered to the Army General Hospital at Fort Bayard, N. M., for treatment, where he died on May 29, 1905.

The death of Assistant Naval Constructor McDonald is a loss not only to our Society, but to the Navy as a whole and one that will be felt by many in other walks of life.

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BOOKS RECEIVED.

A CLASS-BOOK OF NAVAL ARCHITECTURE.—By W. J. LOVETT, Lecturer on Naval Architecture at the Belfast Municipal Technical Institute. New York and London: LONGMANS, GREEN & COMPANY. Price, $2.50.

This book is intended primarily for use in the class room, for which purpose it is well adapted. It will also be found of much value for reference by the professional man as well as the student.

Beginning with a chapter on mathematics, the various subjects relating to ship construction are treated in detail in a brief but clear manner. The work will undoubtedly be in much demand in the various technical schools.

STIRLING.—A book on Steam for Engineers, edited by the engineering staff of the Stirling Company.

This volume contains much valuable information with regard to boilers, arranged in convenient form.

The Stirling boiler is described in detail and a large amount of data given concerning this type. The treatise is subdivided into subjects as follows:

The Stirling Water-Tube Safety Boiler, Water-Tube *vs.* Fire-Tube Boilers, Works of the Stirling Company, Heat, Air, Water, Impurities in Boiler Feed Water, Steam, Moisture in Steam, Flow of Steam through Pipes and Orifices, Superheated Steam and the Stirling Superheater, Combustion, Fuels for Steam Boilers, Determination of Heating Values of Fuels, Fuel Burning, The Stirling Chain Grate Stoker, Utilization of Waste Heat, Chimneys and Draft, Analysis of Flue Gases, Steam Boiler Efficiency, Horsepower Rating of Boilers, Rules for Conducting Boiler Trials, Tables of Tests on Stirling Boilers, Boilers for Mining Service, Principles of Steam Piping, Boiler and Steam-Pipe Coverings, Boiler Cleaning, Care and Management of the Stirling Boiler, Specifications for Masonry in Stirling Boiler Settings.

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ASSOCIATION NOTES.

PRIZE ESSAY.

At a meeting of the Council held on Saturday, March 11, 1905, at the Navy Department, the votes of the members for the prize essay were opened and counted.

As a result of this count, the gold medal and prize of the Society for the most important contribution to the JOURNAL during the year 1904 was awarded to Ernest N. Janson, M. E., for his article entitled "Steam Turbines, with a Special Reference to their Adaptability to the Propulsion of Ships," which appeared in No. 1, Vol. XVI.

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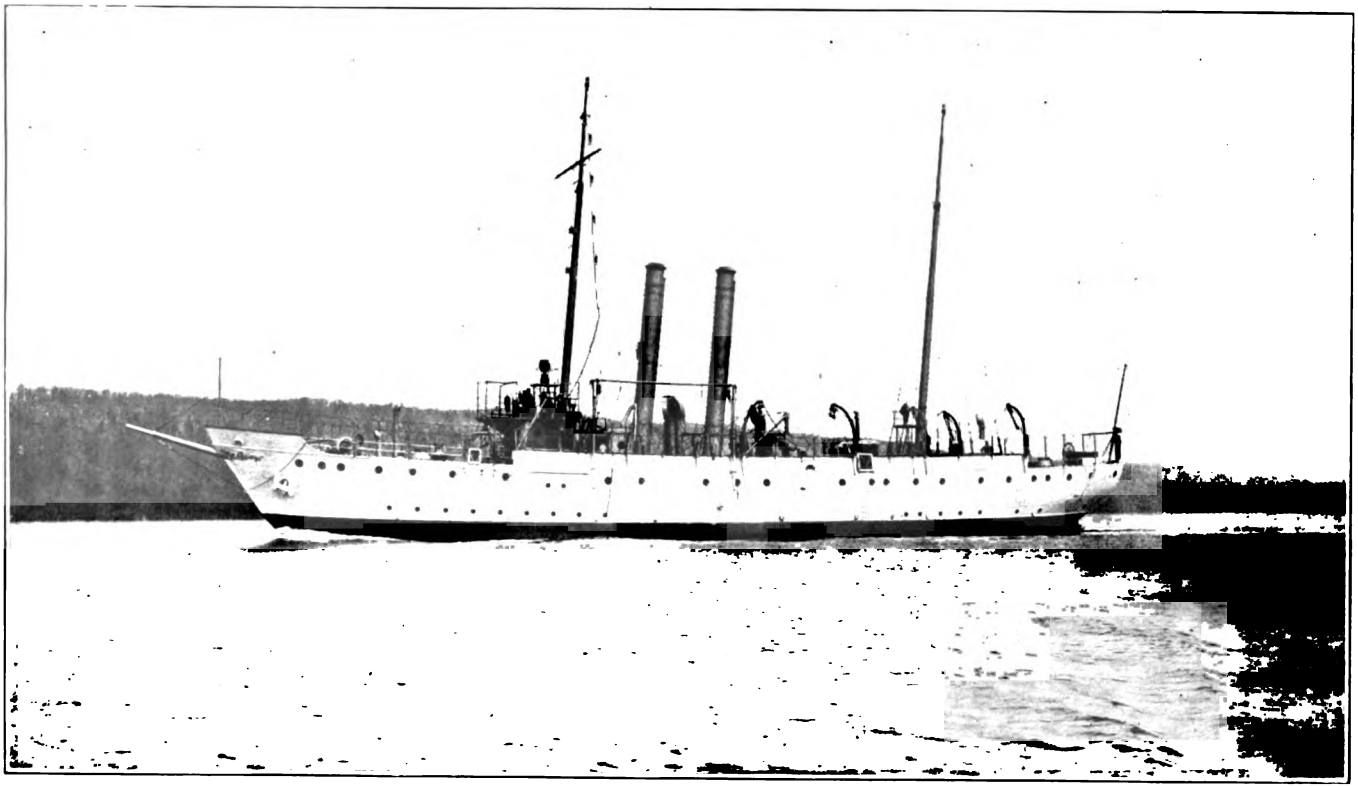

U. S. GUNBOAT "DUBUQUE."

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The Society as a body is not responsible for statements made by individual members.

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(Under whose supervision this number is published).
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U. S. GUNBOATS *DUBUQUE* AND *PADUCAH*.

BY COMMANDER W. N. LITTLE, U. S. NAVY, MEMBER.

DESCRIPTION.

An Act of Congress making appropriation for the naval service, approved July 1, 1902, authorized the construction of two 12-knot gunboats of about 1,000 tons each, all parts of which were to be of domestic manufacture.

Bids for the construction of both vessels were duly advertised, and the contracts for their construction awarded separately to the Gas Engine and Power Company and Charles L. Seabury & Company, Consolidated, of Morris Heights, New York.

The contract for gunboat *No. 17*, the *Dubuque*, is under date of May 29, 1903. For gunboat *No. 18*, the *Paducah*, the sister ship, the contract was not signed until July 6, 1903, after re-advertisement.

The contract price for the *Dubuque* was two hundred and ninety-five thousand dollars. For the *Paducah* it was three hundred and fifty-five thousand dollars.

This price was for hull, machinery and equipment, excepting stores, and provided for the installation of the armament

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and ordnance outfit, complete, by the contractors after it is supplied by the Government.

The contract time for completion of the *Dubuque* was twenty months; afterwards extended, because of difficulty in obtaining materials and because of strikes and labor conditions in vicinity of works, six months, to June 1, 1905.

For the *Paducah* the contract time for completion, twenty months, was extended four months, to July 6, 1905.

As these vessels are, one built and the other building, from the same plans and specifications, prepared by the Navy Department, a description of one should suffice for the other.

The *Dubuque* is a composite-built gunboat of 1,080 tons displacement, with two masts, two funnels, flush main and gun decks and an interrupted berth deck. The principal dimensions and characteristics of the hull are as follows:

PRINCIPAL DIMENSIONS OF HULL.

Length between perpendiculars, feet	174
on L.W.L., feet	174
over all, feet and inches	200- 9
Beam, extreme, feet	35
at L.W.L., feet	35
Ratio of length to beam	4.971 to 1
Depth, molded, main deck, feet and inches	22- 9⅛
Draught, normal, mean, feet and inches	12- 3
mean on trial, feet and inches	12- 3
Displacement, normal draught, tons	1,085.3
trial, tons	1,085.3
per inch at L.W.L., normal draught, tons	10.51
Area midship section to L.W.L., normal draught, square feet	348
L.W.L. plane (draught 12 feet 3 inches), square feet	4,470.7
Wetted surface, square feet	6,900
Center of gravity of L.W.L. plane abaft M.S., inches	3
buoyancy above bottom keel, feet and inches	7- 3
forward of M.S., feet and inches	2-11¼
Transverse metacenter above center of buoyancy, feet and inches	9- ⅞
Longitudinal metacenter above center of buoyancy, feet	194
Coefficient of fineness, block	.509
of M.S	.812
L.W.L	.732
Cylindrical coefficients, longitudinal	.627
vertical	.904
Number of frames	116
Number of compartments	51

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ARMAMENT.

Six rapid-fire 4-inch guns; four rapid-fire 6-pounders; two rapid-fire 1-pounders; two Colt automatic.

The ship is propelled by twin outward-turning screw propellers driven by a pair of three-cylinder triple-expansion engines. Steam is generated by two Babcock & Wilcox marine boilers under a gauge pressure of two hundred and sixty-five pounds per square inch, intended to be reduced to two hundred and fifty pounds (maximum) at the engine.

No armor or protective decks are present.

HULL.

The above water form of the hull is unique in appearance for a naval vessel although not uncommon in a steam pilot boat or light ship. The stem slopes well aft below the water line and the heel is reduced or cut away for ten feet from stern post to facilitate turning and to mimimize the tactical diameter.

The flaring bow affords excellent deck space for working the anchor gear and should be advantageous in a head sea.

The rudder, which is of bronze, of the barn-door type, is suspended by three pintals on its forward edge and its lower edge is thirty-three inches above the lowest part of the keel and is about in line with the heel. Its area is fifty-two square feet.

The frames of the hull above the floors are fifteen-pound channel bars, spaced twenty inches apart, excepting near the bow and the stern.

The vertical keel is thirty inches deep between inner and outer bottoms, lightened by oval holes. The depth is uniform between frames 28 and 100, decreasing forward. Under the engine room, seven feet each side of the keel plate, are longitudinals between inner and outer bottoms. On the starboard side the longitudinal girder merges into the bunker bulkhead. On the port side it ceases at the fireroom; the port bunker (fireroom) bulkheads extending to the outer bottom. Additional intercostal girders support the inboard

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seatings for the engine bed plate. Below and for twenty-six inches above the water line the hull is sheathed and coppered. Excepting for a small space above the bilges the hull is plated overall, the plating being about fourteen pounds to the foot. Under the boilers and magazine spaces are water bottom compartments, seven in all, having a capacity for forty-eight and eight-tenths (48.8) tons of fresh water. Four of the compartments under the boiler compartment are fitted for reserve tanks, and have an average capacity each of four and thirty-five-hundredths (4.35) tons of fresh water. These reserve tanks can be pumped from one to another. There is a trimming tank forward, of 525 cubic feet capacity and one aft of 214 cubic feet which can be pumped through the secondary drain, and filled from the sea or through hose and manhole at top.

The planking is from four and one-half to three-inch yellow pine.

The berth and gun decks are steel throughout, covered with linoleum. The main deck is stiffened by stringers, waterways and plating in the waist and planked overall.

On the main deck are the pilot and chart house, over which is the bridge, equipped as customary with electric and mechanical annunciators to engine room and an electric revolution indicator from each engine shaft of the vibrating-arrow type. Abaft and contiguous with the pilot and chart house is the galley, followed by the boiler-room hatch and two funnels, an athwartship passage, a wireless-telegraphy house, the engine-room hatch, an athwartship passage, hatch to berth deck, ward-room skylight, standard compass platform, main mast, hatch to "cabin country," and, finally, the hand-steering gear.

The gun-deck space is divided and occupied as follows, the spaces considered in order from forward:

A paint room in the eyes of the ship contains numerous tanks (nine), with glass gauges, for oil; others (five) for mixed paints and dry materials. It is lighted, as is the gun deck, throughout by air ports and dead lights of ample size, and is well fitted with shelves. Abaft the paint room is a midship passage flanked by the crew's water closet on the starboard

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side and the crew's wash room on the port side. In the passage are valves controlling the head-flushing system, three spigots for salt water from same pipes and a locked spigot for fresh water. The crew's lavatory contains seven tilting and removable glazed basins, each under a spigot and over a trough, and four shower compartments, with hot-water facilities and a spigot for bucket filling.

In the next compartment, the second from forward, are the steam capstan engine, worm wheel and vertical shaft, and two chain compressors of the ordinary type; on the starboard side, the ship's prison and scuttle butt; on the port side, a storeroom and a large ice chest; next are two 4-inch mounts and ports, one each side. Under the pilot house is the armory.

In the third compartment are, amidships, the firemen's washroom, with six basins and spigots, two shower baths, and a fresh water tank of 300 gallons capacity, tiled floor, two mounts each side for 6-pounders, the fireroom hatch way and drum rooms, in the wake of which, amidship, is the machine shop, containing a lathe, shaper, drill press and a tool grinder driven by an electric motor; also a small bench with two vises.

Abaft the machine shop is the engine-room hatch trunk, extending, as do all principal bulkheads, from ship's bottom to the main deck, abaft of which is the main hatch to main and berth decks. Abreast those hatches are the following rooms: starboard side—pay office, executive officer's office, executive officer's room; port side—engineer's office, ward room officer's lavatory, bath room and water closet.

Next occurs bulkhead No. 82, abaft of which is the wardroom officers' mess-room amidship and on port side, pantry amidships, flanked and succeeded by eight state rooms and a passage. The captain's cabin, aft of all, extends from side to side abaft bulkhead No. 105, and is about 15 feet wide by 13 feet long. Off and forward of the cabin are one stateroom on starboard side and cabin pantry on port side. In the overhanging stern and over tiller room is the captain's lavatory, with bath and water closet of the yacht type, with

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tiled floor. The 'tween-deck distance is six feet three inches in the clear.

The berth deck is interrupted by the machinery spaces with neither bulkhead doors nor passages to connect fore and aft.

In the bows over the trimming tank are the sail room and equipment store room to frame 17; crews' space to frame 45, with an intermediate watertight bulkhead at frame 27. In the forward of these two berth-deck crew spaces, *i. e.*, compartment A–11, are located three 120-fathom hawser reels, leads to chain lockers and hatches to store room below. In the second crew space, compartment A–12, are the hatches to fore hold and forward magazine, the steering-engine and dynamo-engine room.

After berth-deck spaces from engine room are the dispensary, sick bay and petty officers' and marines' quarters; abaft of which space, with a watertight bulkhead and door between, is the mess attendants' quarters, three store rooms and tiller room. The absence of a steering engine and dynamo engine from aft affords unusual space for crew and stores in this part of the ship. The tiller arc or quadrant fits closely under the gun deck, occupying very little space.

The sick bay has six beds of the folding or close-stowing type, no woodwork in vicinity excepting oak desks and drawers in dispensary. All light bulkheads are of corrugated galvanized iron. A ventilating electric blower of 2-horse-power is located in a recess on starboard side of after berth-deck passage, *i. e.*, in the mess attendants' quarters.

Below the berth deck, forward of the machinery spaces, occur the trimming tank, dry provision, general and construction store rooms and the forehold (with cable tier) as far as frame 33. In this hold outboard, forward of and adjacent the forward magazine, are the two main fresh-water tanks. Between these tanks in separate compartments are the electrical and ordnance stores. The forward magazine is situated between bulkheads at frames 36 and 45. The 6 and 1-pounders are stowed outboard and the 4-inch amidship. Between the for-

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ward magazine and boiler space the bulkhead is sheathed with asbestos millboard and an air space is provided. A similar provision is made for the protection and ventilation of the forward bulkhead of the after magazine in wake of the engine room.

Below the berth deck abaft the machinery spaces, on the port side of the shaft alley, are the paymaster's small stores and petty officers' stores. On the starboard side, and entered from the starboard-shaft alley, are the machinery stores. In the wake of these store rooms are the small-arms and 4-inch magazine and handling room as far as frame 92, abaft of which are the navigator's stores, flanked by hospital stores on port side and marines stores on starboard side, to frame 98. Paymaster's stores occupy the remaining space aft to the trimming tank.

BOATS.

The boats carried are:

	Capacity, men.	Pounds water and provisions.
One 28-foot steam cutter, . .	35	350
One 28-foot cutter, ...	40	400
Two 26-foot cutters, each . .	30	300
One 28-foot whaleboat, . .	23	230
One 28-foot gig whaleboat, .	16	160
One 18-foot dinghy, . . .	10	100
Two 12-foot punts.		

COMPLEMENT.

Commander, ...	1
Wardroom officers (staterooms), . . .	8
Engine-room force, ...	29
Special branch, ...	11
Seaman branch, ...	73
Artificers' branch, ...	10
Messmen, ...	14
Marines, ...	12
Total, ...	158

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The *drainage system* includes a 7½-inch galvanized, lap-welded steel main drain ¼-inch thick, with terminal suctions in after end of fireroom and engine room, amidships of, and a secondary drain of 4-inch galvanized lap-welded steel pipe $\frac{3}{16}$-inch thickness extending from forward end of the boiler compartment to the after engine-room bulkhead. Secondary drain manifolds are located—one (No. 1) in starboard passage at forward end of boiler compartment, with leads to fore holds and chain lockers, and one at the after engine-room bulkhead nearly amidships, with leads to the shaft alleys and afterbody bilges. Valves controlling the suction forward and aft of the machinery spaces are of the stop-check type, and operated on berth deck, except in shaft alleys, which are operative at the valve only by handwheel below floor plates in engine room. The main drain is connected by a 5½-inch galvanized steel pipe, with both main circulating-pump suction pipes. This connection is made operative by first closing the main injection valve, which act releases and permits the opening of the bilge suction valve at the circulating pump in question, and then opening the 7½-inch special stop check valves operated on gun deck or above floor plates near after bulkhead amidships, one in engine room and one in boiler compartment. There are no Macomb strainers on the main drain. Macomb strainers and 4-inch valves in the secondary drainage system in machinery spaces are three in number, located at the bilge wells, one amidships at the after end of engine room and one about 5 feet to starboard of amidship line between after boiler and bulkhead, one 3-inch branch suction valve and strainer at forward end of engine room. The engine crank pits are drained by connections with the independent fire and bilge pump in engine room and by the main bilge pumps attached to engines.

The double bottoms connected with the drainage system are A–99 and B–95. Compartment B–95 only can be flooded from the sea by a connection at sea suction valve in fireroom. The main drain is pumped only by the main circulating pumps. The secondary drain is pumped by attached engine bilge pumps,

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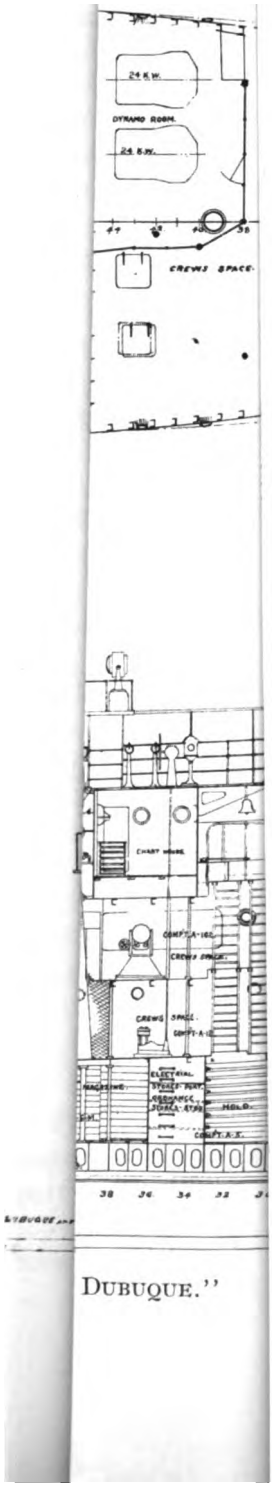

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by the fire and bilge pump starboard side of engine room, by the auxiliary, feed, fire and bilge pump in fireroom, and by two hand pumps on gun deck.

Manifolds on the drainage system are as follows: No. 1. Forward end of boiler compartment, starboard side; No. 2. At auxiliary feed, fire and bilge pump in fireroom; No. 3. Fire and bilge pump in engine room; No. 4. At after bulkhead in engine room.

FIRE-MAIN SYSTEM.

The 4-inch copper fire main starts from the riser at the fire and bilge pump, in engine room and the forward lead passes through the boiler room on the starboard side under the gun-deck beams.

There is a 2½-inch riser from the auxiliary feed pump in boiler room to the fire main. By closing the gate valve in the main near this connection, the fire main is divided into two systems, one forward and one aft. The "auxiliary feed pump" is connected with the former and the "fire and bilge pump" with the latter.

The distiller circulating pump may also be used on the fire main, through a connection between frames 67–68, in engine room.

There is a connection to the sanitary system between frames 19–20, gun deck, forward.

A relief valve, set at 100 pounds, between frames 29–30, discharges into main deck scupper on starboard side of berth deck.

Cut-out valves are situated at the foot of each riser. Connections for flushing ash chutes are taken from this main.

FRESH-WATER SYSTEM.

The fresh-water tanks in hold may be supplied from fresh-water boat by connections on ship's sides between frames 28–29, port and starboard. From these tanks the reservoir tank on top of galley house can be filled by means of hand pump on forward berth deck.

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The supply from distillers may discharge into tank in firemen's wash room and ship's tanks.

The main pierces the gun deck between frames 42–44 from below, leading along under the main deck beams to 80 and 81, at which point it drops to berth deck with branches to pantries and bath rooms.

THE FLUSHING SYSTEM.

The flushing main is connected to the sanitary pump on port main engine. There is also a connection to the distiller circulating pump between frames 69–70 in engine room.

The forward lead passes through the boiler room on the port side, and, continuing along the berth deck, pierces the main deck between frames 20–21, to supply crew's head.

The after lead extends under the gun deck beams, with branches to pantries and bath rooms.

A relief valve, set at 15 pounds, is located in crew's water closet, with discharge into trough.

The flushing water may also be taken from the salt-water discharge from the distillers.

THE VENTILATING SYSTEM.

The ventilating system provides artificial ventilation for all living spaces below the gun deck, including engine room, store rooms, magazine passages, also the air-casing spaces between gun deck and boilers, and between engine and boiler compartments and the magazine adjacent to each; also to the water closets and toilets on the gun deck. As this class of vessels has a high freeboard and hatches opening to the main deck in vertical lines, an exhaust in addition to a supply system was not, excepting for the compartment in which the dynamos are located and the bunkers, deemed necessary. There are six electrically-driven centrifugal fans arranged in as many units or sections of piping. None of the principal bulkheads are pierced by the air ducts excepting at No. 45, where there is a butterfly valve, and the duct supplying air to the double space under gun deck over boilers and to bunkers is of heavy iron pipe. Air ducts connect the bunkers with

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the annular space around uptake and smoke pipes without gates or valves.

The following is a list of the fans by their number, designation and the purpose of each:

Fan No. 1. Exhausts from water closets and wash rooms forward of bulkhead No. 23 on gun deck.

Fan No. 2. Exhausts from officers' water closet and bath rooms on gun deck and discharges into engine-room hatch.

Fan No. 3. Exhausts from dynamo-room compartment and discharges into air space over boilers.

Fan No. 4. Supplies fresh air to dynamo room and to all compartments forward of bulkhead No. 45 and below gun deck.

Fan No. 5. Supplies fresh air to engine room.

Fan No. 6. Supplies fresh air to all compartments abaft of bulkhead 79 and below gun deck.

PARTICULARS OF EACH VENTILATING FAN.

Fan No.	Supply or exhaust.	Size.	Rated power.	Cu. ft. air delivered per minute.	Kilowatts absorbed.
1.	Exhaust.	No. 2 Monogram.	½ H.P.	570	.51
2.	Exhaust.	No. 2 Monogram.	½ H.P.	570	.49
3.	Exhaust.	30-inch.	1.5 H.P.	1,612	1.53
4.	Supply.	45-inch.	3.5 H.P.	3,815	3.35
5.	Supply.	50-inch.	4 H.P.	4,625	4.23
6.	Supply.	35-inch.	2 H.P.	2,040	1.83
Total.....	...	.....	...	...	11.94

STEERING GEAR.

The steering gear consists of a steam steering engine located in compartment A-12 (berth deck, forward), under the chart house and steering wheel, connected by steel-wire rope passing along under the gun deck to the rudder arc and a reverse-screw hand steerer located at the rudder head on the main deck.

The steering engine is an inverted twin-cylinder direct-acting steam engine with a float valve, driving a worm on the crank shaft which engages a worm wheel on a vertical shaft,

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which shaft carries a pinion engaging a spur wheel on a second vertical shaft. This latter shaft has fixed near its upper end a two-grooved pulley, to which one end of each tiller rope is fixed so as to unwind one rope while winding the other. Outboard are leading sheaves adjustable for varying the tension of the tiller ropes. The tiller ropes are led tangentially and fixed to the tiller quadrant by pelican hooks, which latter must be detached when the hand steerer aft is shipped. There is a clamp at the rudder head for use while making the change from steam to hand gear and vice versa. Before reconnecting the tiller ropes to the tiller arc it may be necessary to slacken the ropes at the guide sheaves aforestated. The links of the hand steerer are of the sort in common use in the service, where the reverse-screw shaft is worked by an engine or by handwheels, and are thrown in and out of gear by hand and secured by removable pins in the usual way.

THE CAPSTAN.

There is one capstan forward of pilot house on main deck, driven by a twin-cylinder, simple, vertical engine, 7 inches by 8 inches, located nearly underneath, on gun deck, through a worm and worm-wheel gearing and vertical shaft. Under the capstan drum is a wildcat fitted with a band brake for taking the chain cable, which wildcat can be operated either by hand, through the capstan or be thrown out of gear. The capstan above the wildcat can be operated by hand, with capstan bars, or by the engine, by the adjustment of clutch keys for the purpose. There are two large chain bits forward and four leading bits on the capstan bed plate.

The bow anchors stow in the hawse pipes and weigh, 3,000 pounds each.

ELECTRIC EQUIPMENT.

The electric generating installation consists of two multipolar continuous-current generators each driven by a compound engine, both manufactured by the General Electric Company. Engine marks: type, CC-24-400; form, H-4. K.W. 24; steam, 100, revolutions, 400. They are vertical direct-acting, and have

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cylinders of 6½ and 10½ inches diameter by 7 inches stroke. Diameter of piston rods, 1 $\frac{3}{16}$ inches. The starboard engine bears shop No. 2,228. The port engine bears shop No. 2,229.

The dynamos are of type, M.P.; class, 8-24-400; form, D; volts, 125; amp., 192; K.W., 24; speed, 400; shunt current, 4.1. The starboard dynamo bears shop No. 87,344; the port dynamo bears shop No. 87,345.

All the wiring is in conformity with U. S. Navy standards, in wrought-steel enameled-pipe conduits on the two-wire feeder system.

There is one searchlight located on a platform over and abaft the bridge. Its projector is 18 inches diameter. Other lights distributed through the ship as follows:

Berth, gun and main decks forward are lighted by twenty-four 16-candlepower lamps.

Berth, gun and main decks aft are lighted by fifty-nine 16-candlepower lamps.

The engine room is lighted by thirty-two 16-candlepower lamps including six portables. (Additional lights are being installed.)

The fireroom is lighted by twelve 16-candlepower lamps including three portables.

There are forty-four battle lights of 16-candle power, and twelve of 32-candlepower.

MAIN ENGINES.

There are two *main engines* located in one compartment abreast one another. They are vertical, three-cylinder, triple-expansion, direct-acting, with air pumps, main-feed, sanitary and bilge pumps attached. The original plans contemplated inward-turning screws, which were changed to outward-turning by exchanging the positions of the engines, shafting and propellers. This new arrangement placed the starting gear on the outboard sides and far apart, necessitating two men for working engines to signals.

The cylinders are of cast iron, unjacketed, the H.P. cylinder forward, and rest on wrought-steel cylindrical columns

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and cast-iron box columns. The former columns are outboard, extending from cylinders to bedplate; the latter are bolted between the housings on top of condensers and lugs on cylinders. Brackets on the inboard columns support the reversing shafts. The cylinders and steam chests, including the bottom heads, are cast iron, one piece—the H.P. forward—and are bolted together. Steam enters the H.P. chest between the valve heads, and exhausts at their ends, passing around the H.P. cylinder in passages through the casting to the ends of the intermediate valve chest, exhausting from the middle through a 7-inch copper pipe to the L.P. steam chest.

The main pistons are made of cast steel, class A, and, with their followers, are finished all over. The followers are wrought steel. They are fitted with two packing rings each, of cast iron, tongued where cut obliquely, and retained in a fixed relative position by the heads of the bolts securing the tongue pieces engaging slots in projections on the pistons. No restraining lugs or distance pieces are provided, nor are any springs provided for setting out the rings.

The engine bed plates are of cast iron, each in one piece, resting on three longitudinal seatings worked in the ship's structure on top of the ship floors, and two intercostal girders. No diagonal braces are provided for these engines, as the rigid construction of the condenser and inboard columns seems all sufficient for the small masses involved.

The main valves are one for each cylinder, of cast iron, and of form as follows: For the H.P. and I.P., single-ported piston valves with equal heads; for the L.P., flat double-ported slides. No balancing is provided except by a gravity-balance steam piston working off the low-pressure valve, the top connected permanently with the main condenser by a pipe. All main valves are operated directly by Stephenson double-bar links and eccentrics. There is no independent cut-off gear other than adjustable connections between the reach rods and reversing-shaft arms. The slots in the reversing-shaft arms are graduated by marks indicating the mean of the up-and-down points of cut-off in inches from beginning of stroke.

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Main Valve Packing.—The intermediate piston valves only are fitted with packing rings, and suitable lugs are provided to prevent them from overriding the seats when the valves are disconnected. These packing rings are in one piece for each end of the piston valve with a single parting for expansion and adjustment. At the parting are lugs with bolts and jam nuts. All the main H.P. and I.P. valve chests are fitted with cast-iron liners. The cylinders have no liners.

The piston and valve rods are packed with United States metallic packing of latest design, consisting of a combination of beveled white metal rings and spiral springs. It was fairly tight during the trial.

The crank shafts are solid class A forgings, excepting an axial hole through each crank pin, and are each in one piece. The cranks are set at 120 degrees apart. Between the intermediate and L.P. cranks is a worm wheel for turning the shaft by hand gear. There are six crank-shaft bearings, one pair to each crank, with two bolts each. Each bearing is composed of a bottom brass, two distance pieces and a cap. The cap is of cast steel, class A, the other parts of composition. The cap and brass are lined with white metal and provided with oil channels, into which the oil enters near the side.

The piston rods are of high-grade steel forgings, solid, accurately ground and polished. The upper end of each is secured to the piston by a shoulder, taper, nut and split key. A dowel pin between piston and rod prevents turning between the parts. The lower end of the piston rod includes the crosshead, to which is bolted a composition slipper of manganese-bronze, white-metal faced both sides. The end of the piston rod or gudgeon carries a pair of brasses, white metaled, with a two-bolt cap for the connecting-rod pin. All crossheads, piston and connecting rods are of the same dimensions for the different sections, *i. e.*, cylinders.

The connecting rods are of high-grade steel forgings, forked at one end to span the crosshead, the crosshead pin shrunk in, secured from turning by screw keys and fitted with white-metaled crank-pin brasses at the other. The lower end of the

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connecting rod is flat, against which are bolted the crank-pin brasses and steel binder by two high-grade steel bolts. The intermediate crosshead journals or pins differ from the others in the presence of wrist pins for the air-pump beam connecting links. For one-third of their projected area the crosshead pins are flattened on their (inboard and outboard) sides.

The eccentrics are of cast iron, each in two parts bolted together, rabbetted at each side for the flanges of the eccentric straps. Each backing eccentric is secured to the shaft by a key and each go-ahead eccentric likewise, with additional adjusting pieces between its key and the eccentric.

The eccentric straps are of composition, lined with white metal, which latter is dovetailed and hammered in place. The eccentric rods of forged steel have forked ends at their link connections and T-shaped heads where bolted to the eccentric straps. The link wrist-pin brasses are adjustable.

The reversing gear for each engine consists of a steam cylinder 5 by 9 inches secured to the intermediate cylinder casting, a reversing shaft resting in bracket bearings on inboard engine columns, an eccentric secured to the reversing shaft, connecting and reach rods, and manipulating lever with locomotive latch at working platform. The reversing-engine piston is of cast iron fitted with cast-iron snap rings. The reversing-engine valve is a plain D slide, which receives its primary motion from the reversing hand lever and its secondary motion from an eccentric on the reversing rock shaft. No hand-reversing gear is provided. The engines are fitted with a hand turning gear consisting of an adjustable steel worm on a short shaft with square end, engaging a worm wheel on the crank shaft between the intermediate and low-pressure cranks. A ratchet wrench working horizontally affords the necessary purchase. The worm is removable by means of a two-bolt steel binder.

The *main condensers* are of the surface-condensing type, steam outside the tubes, with shell of cast manganese-bronze of rectangular form in horizontal section side elevation and plan with cylindrical tops, flat bottoms and sides. The side of

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each condenser nearest the crank shaft is inclined sufficiently to accommodate the crank circle. They are slightly longer than the engines, projecting aft beyond the engines only by the by-pass channel and valve on the after heads, and form the housings for the inboard sides of the main cylinders. On the tops and integral with the shells of each condenser are the following nozzles: Circulating-water outlet, exhaust steam, safety valve and two 6 × 4 handholds; also on top are bosses for reverse-gear brackets and housings for the air, feed, bilge and sanitary-pump rock-shaft pillow blocks. The thickness of the shell is seven-sixteenths of an inch, stiffened by rectangular ribs where necessary. A division plate at the after ends causes the circulating water to pass forward through the lower nest of tubes and aft through the tubes of the upper nest. For the purpose of pumping from the bilge by the main circulating pumps a by-pass valve and pipe of rectangular section connects both sides of the division plate. The auxiliary exhaust and the bleeder pipes are connected with the main exhaust pipe. There is a connection of four inches diameter for the air-pump suction and one two and one-half inches diameter for a pipe between the condensers at bottom. Joining this latter pipe is a permanent steam connection with the bleeder-pipe cross, one inch in diameter, for boiling out the condenser. The lower valve on this connection is under the floor plate amidships in line with both circulating pumps. A horizontal perforated brass plate forms a diaphragm within, completely across and five and three-quarter inches from the top of the condenser, leaving a clear space over the ends to scatter the exhaust steam. The perforations, $\frac{1}{2} \times \frac{3}{4}$ inch pitch, occur for forty-seven inches by seventeen and a quarter inches forward of the exhaust nozzle; under the exhaust nozzle the plate is not perforated. The tubes, in two nests are straight, supported by a middle diaphragm and packed in rolled tube plates of same composition as tubes by screwed glands over cotton tape. The condensers were tested under thirty pounds hydrostatic pressure, and the safety valve afterwards set at 25 pounds.

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Particulars of condensers.

Number of condensers	2
Steam inside or outside tubes	Outside.
Cooling surface, square feet, starboard	678.716
port	678.716
total	1,357.432
Ratio of total cooling surface to total heating surface	1 : 3.0638
Number of tubes to each condenser	488
Diameter of tubes, outside, inch	$\frac{5}{8}$
Length of tubes, feet and inches	8– 6
Spaced, inch	$\frac{15}{16}$
Composition	Cu. 70, Zn. 29, Sn. 1.
Thickness of tube plates, inch	$\frac{7}{8}$
Ferrules for packing, outside diameter	$\frac{7}{8}$-inch, 18 threads.

An air pump located inboard of each condenser is worked off the intermediate crosshead of each engine, as also are sanitary, bilge and main feed pumps, arranged as follows:

Forward of both air pumps and attached to arms on the main-pump rock shafts are the feed pumps; abaft the air pumps are the bilge and sanitary pumps—the former to the port engine, the latter to the starboard engine. The bilge, sanitary and feed pumps are of the differential plunger type with water-packing grooves. All pumps on main engine and the main and auxiliary feed, fire and bilge pumps have spring safety relief valves. The air pumps are of the vertical-bucket closed-top type with sheet-bronze valves. All these engine pumps are of composition of the Navy standard in their chambers, buckets, plungers and valve chests. Each pump has an independent crosshead attached by double-link connections of steel rods and composition bearings to the pump rock-shaft arms. The air-pump guides are bolted to and the other pump guides are cast with their respective top bonnets. The main-pump rock shaft is supported by the pillow blocks, of two parts each, straddling each pump arm, resting on housings cast on the top outboard side of each condenser.

The thrust, *stern sleeve* and *propeller shafts* are each in one section, constituting with the crank shafts the main shafting of the vessel. They are all (excepting the crank pins) of solid class-A forgings, finished all over to a diameter of 5 inches

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before encasing. The propeller, stern-tube and thrust-shaft after couplings are double-taper steel sleeves, with two "feathers" or longitudinal keys, 90 degrees apart each. Two through keys at each sleeve coupling, one through each section end, bind the sections together. At these couplings the ends of the shafts are one-half inch apart. Screw keys between the through keys and sleeves secure the keys in place. Five-bolt flange couplings unite the crank and thrust-shaft sections.

The stern-tube and propeller shafts are encased by composition sleeves, pinned on with brass stop screws, spaced spirally. A fair water containing pitch encases the outboard shaft couplings. Each thrust shaft has five thrust collars.

There are two outboard brackets or struts and hangers to each shaft, of manganese-bronze, with sleeves lined by lignum-vitae strips, end of grain to the shaft. The stern and bracket sleeves are in halves longitudinally. The stern tubes have a bearing at each end.

The *thrust bearings* each consist of a pedestal and bearing proper. The bearing is fitted in a bored part of the pillow block, which latter is bolted to its seating, built in the ship's structure. It is in two parts, of cast iron, with white metal linings. The upper part is fitted to the lower with dowel pins and longitudinal recesses in the upper flanges of the pedestal, to which it is held down by steel bolts, body bound in the pedestal, and with slotted holes in the cap. The ends and side walls of the pedestal form an oil trough, from which lead oil holes to the collars and recesses. Stuffing boxes and glands are provided to retain the oil, and four adjusting screws in lugs on the pedestal provide longitudinal adjustment of the bearing. Forward of each thrust bearing on the same pedestal is a spring bearing, resting on and adjustable vertically by a broad wedge liner. This wedge is between the spring-pillow block and base of thrust-bearing block, adjustable by set screws on each side. A mandrel is provided for rebabbitting the thrust bearing.

Main Circulating Pumps.—Two compound engines drive two centrifugal pumps—one each side of engine room. The engines are very light, both cylinders and valve chests in one

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casting, with forged finished-steel column supports, cast-iron bed plate, valve rock shaft, cranks at right angles, complete adjustability of all parts and fitted with indicator gear and counters. The H.P. piston is of cast iron, fitted with two snap rings, without follower, of cast iron. It is screwed on the piston rod, the end of which is riveted over in a countersunk recess in piston. The L.P. piston is of cast iron fitted with two rings and a follower. The piston and follower are secured to the rod in the manner described for the H.P. piston with the addition of rivets through piston and follower instead of follower bolts. The H.P. valve is of the solid-piston type. The L.P. valve is a D slide with a composition spring-borne tongue fitting into its back, as a guide and to press the valve against its seat.

The pumps are two, one to each engine, of composition throughout. Each runner has eight blades, curved at the tips. The runner is keyed by a tobin-bronze key to a tobin-bronze shaft. A four-bolt flange coupling joins engine and pump shafts. The casing is in two parts joined together by a nearly horizontal flange joint and brass bolts. The lower half has, at and near its base, three nozzles, one pair connecting with the sea injection and with the main drain with interlocking valves; the other nozzle connecting with the circulating water pipe to the condenser. Suitable channels in lower section of casing lead the suction to both sides of the runner at center. There is a lignum-vitae bearing, covered by a composition circular plate, for the after end of the runner shaft. A stuffing box and gland only is provided for the forward end, as the engine shaft bearings are adjacent.

CIRCULATING PUMP DATA.

Diameter of H.P. cylinder, inches	3
L.P. cylinder, inches	6½
Stroke, inches	5
Diameter of piston rods, inch	⅝
pump runners, inches	18
Width of pump runners, inches, at periphery	2¾
hub	4¾
Diameter of inlet nozzle, inches	5½
outlet nozzle, inches	5½

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There are two *feed and filter tanks* of ⅛-inch galvanized sheet steel in the engine room, one forward of each engine. The capacity of each is 245 gallons, about, including filter boxes. They are clothed with hair felt and sheathed by planished iron. The filter compartments, four in number, occupying about one-fourth the total of the tanks, are worked in the upper part of the tank with partitions watertight at the sides and bottom, such as to cause the water from the air pumps to flow through filter material from top to bottom of each filter compartment, passing between them from the bottom of each to the top of the next. Each tank is provided with a safety relief valve on the first filter compartment and a vapor pipe from both the first filter compartment and side near top of feed tank. The top and removable cover are of composition. At the inboard end of the top of each tank are four nozzles or openings for vapor pipe to deck, spring relief valve, connection from cylinder relief valves and for air-pump discharge. The vapor pipe and cylinder relief-valve pipe openings are 2½ inches diameter; the air-pump discharge and the cylinder relief valve to tank are 3-inch diameter each. The spring relief valve from tank discharges into a 3½-inch overflow pipe with open funnel top to bilge. At the bottom of each tank is a ¾-inch drain cock. A 3-inch suction opening to feed pumps at bottom inboard side of each tank is controlled by an automatic ball float valve, or may be kept open by hand lever and latches to valve mechanism. All filter compartments can be drained into lower reservoir by suitable ½-inch valves and pipe on the front of tanks. The tanks are connected by a pipe and valve between the two tanks for the purpose of equalizing the water level in both or for using either tank separately. As the main and auxiliary cylinders are not equipped with oil cups, no considerable cylinder oil is likely to be found in the filter tank other than may get into the L.P. cylinders by way of the packing around the piston rods. There are no independent filters on the feed-water system.

The main feed pumps, one worked off each main engine, are of composition of the differential plunger type and have the following dimensions:

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Diameter of plunger, greatest, inches	$3\frac{5}{8}$
least, inches	$2\frac{9}{16}$
Stroke, inches	4.95

The *air pumps* are two, of the bucket plunger type, one worked off each main engine, and have the following dimensions:

Diameter, inches	12
Stroke, inches	6.9
Diameter of pump rod, inches	$1\frac{1}{4}$

The sanitary and bilge pumps attached are of the differential plunger type and have the following dimensions:

Diameter of plunger, greatest, inches	$5\frac{1}{4}$
least, inches	$3\frac{3}{8}$
Stroke, inches	4.95

MAIN ENGINES.—PARTICULARS OF ONE ENGINE.

Cylinders, number for each engine	3
H.P., diameter, inches	9
I.P., diameter, inches	$15\frac{1}{2}$
L.P., diameter, inches	$25\frac{1}{2}$
Stroke of all pistons, inches	21
H.P. valves (piston), diameter, top, inches	$4\frac{3}{8}$
bottom, inches	$4\frac{3}{8}$
I.P. valves (piston), diameter, top, inches	8
bottom, inches	8
L.P. valves (slide), length, inches	21
width, feet and inches	2- $2\frac{1}{2}$
Valve stems, diameter at stuffing box, inches	$1\frac{5}{8}$
below stuffing box, inches	$1\frac{3}{4}$
through valve, inches	$1\frac{1}{2}$
L.P. valve, stem, diameter at balance pistons, inches	$1\frac{1}{2}$
Main steam pipe, diameter at throttle, inches	3
area of cross section, square inches	7.0686
Volume swept by H.P. piston, mean, per stroke, cubic feet	0.7731
I.P. piston, mean, per stroke, cubic feet	2.2197
L.P. piston, mean, per stroke, cubic feet	6.2065
Clearance of H.P. cylinder, per cent., top	19.83
bottom	19.64
I.P. cylinder, per cent., top	17.44
bottom	13.98
L.P. cylinder, per cent., top	9.75
bottom	6.64
Net area of I.P. to H.P. piston	2.871 : 1
L.P. to H.P. piston	8.0271 : 1

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Piston rods, diameter, inches	$2\frac{1}{4}$
length, feet and inches	3- $6\frac{23}{32}$
Cylinder walls, H.P., I.P. and L.P., thickness, inch	$\frac{7}{8}$
Valve-chest liners, thickness, inch	$\frac{1}{2}$
Cylinder relief valves (one each end cylinder), diameter, inches	$1\frac{1}{2}$
Connecting rod, length, center to center, inches	$52\frac{1}{2}$
diameter, upper end, inches	$2\frac{1}{4}$
lower end, inches	$2\frac{7}{8}$
crosshead bolts (4), diameter, inch	$\frac{3}{4}$
Connecting-rod crank-pin bolts (2), diameter, inches	$1\frac{1}{2}$
Crosshead, surface, ahead, square inches	70
backing, square inches	35
pin, diameter, inches	$3\frac{3}{8}$
length of pin, inches	$5\frac{1}{8}$
Crank shaft, number of sections	1
diameter, inches	5
coupling discs, inches	$10\frac{1}{4}$
coupling bolts, number in one flange	5
diameter, inches	$1\frac{1}{4}$
journals, diameter, inches	5
length, inches	5 and 7
length of sections, feet and inches	one piece.
maximum distance, beyond journals, feet and inches	1- 3
Crank pin, diameter, inches	$5\frac{1}{4}$
length, inches	8
projected area, square inches	42
diameter of axial hole, inches	$2\frac{1}{2}$
webs, width, inches	6
thickness, inches	$3\frac{3}{8}$
Thrust shaft, diameter, inches	$4\frac{3}{4}$ and $4\frac{7}{8}$
length, feet and inches	19- $3\frac{3}{4}$
steady bearings, inches	11
collars, number on each shaft	5
diameter, inches	10
thickness, inches	$1\frac{1}{4}$
space between, inches	$1\frac{7}{8}$
total surface, both engines, square inches	598.75
Line shaft, diameter, inches	5
sleeve coupling, inches	8
length, feet and inches	16-$10\frac{1}{4}$
Propeller shaft, diameter over liners in strut bearings, inches	6
after stern-tube bearings, inches	6
in forward stern-tube bearings over liners, inches	6
length, feet and inches	25- $8\frac{5}{16}$
Shaft-sleeve couplings, diameter, inches	8
length, inches	$20\frac{1}{2}$
Stern-bracket bearings, diameter, inches	6

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Stern-bracket bearings, length, feet and inches	1–11½
Stern-tube bearings, diameter each, inches	6
length, each, inches	30
bracket bearings, forward, diameter, each, inches	6
length, each, inches	17½
after, diameter, each, inches	6
length, each, inches	23½
Reversing engine, diameter of steam cylinder, inches	5
stroke, inches	9

VALVE DIAGRAM DATA.

Type of engines	Direct-acting triple-expansion.
Diameter of cylinders, inches	9, 15¼, 25¼
Stroke of cylinders, inches	21, 21, 21
Valve gear	Stephenson type with double-bar links.
Revolutions per minute	225
Piston speed per minute, feet	787.5

	H.P.		I.P.		L.P.	
Eccentricity, inches	1¾		1¾		1¾	
Travel of valve, inches	3¼		3¼		3¼	
Type, number and diameter of piston valves	One 4¼ inches		One 8 inches		*	
Side of valve on which steam is taken	Inside.		Outside.		Outside.	
	Top.	Bottom.	Top.	Bottom.	Top.	Bottom.
Width of port, inches	1¼	1¼	1¼	1¼	2 × 1¼	2 × 1¼
Length of slide valves, inches	...	...	...	...	24¾	24¾
Steam lap, inches	11/16	13/16	17/16	23/16	1/8	13/16
Exhaust lap, inches	—3/32	3/64	—1/8	1/8	3/64	7/64
Angular advance, degrees	35	35	37	37	39	39
Steam lead (angular), degrees and minutes	7, 57	12, 25	8, 15	12, 45	9, 0	13, 5
(linear), inches	15/64	31/64	13/64	31/64	7/32	31/64
Cut-off, in inches	16.212	15.351	15.666	14.721	15.162	14.070
in decimal of stroke	.772	.731	746	.701	.722	.670
Release, in inches	2.062	1.937	2.156	2 015	1.5	1.718
in decimal of stroke	.097	.093	.102	.096	.072	.081
Compression, in inches	1 66	1.80	2.015	2.156	3.406	2.86
decimal of stroke	.079	.084	.096	.102	.162	136
Steam opening, linear, from diagram, inches	31/32	15/16	33/32	1 1/32	2 × 7/8	2 × 13/16
area of, in square inches	10.74	12.17	18.12	20.05	43.31	48.70
Exhaust opening, linear, from diagram, inches	1¼	1¼	1¼	1¼	2 × 1¼	2 × 1¼
area of, in square inches	16.84	16.84	29.93	29.93	61 87	61.87
Steam velocity through ports, in ft. per minute	4,664	4,116	7,938	6,964	9,285	8,279
Exhaust velocity through ports, in ft. per min.	2,980	2,980	4,806	4,806	6,500	6,500

Velocity through main steam pipe in feet per minute	7,095
Velocity of exhaust to condenser in feet per minute	7,088

* One double-ported slide valve.

Data refers to valves in full gear "ahead" and full gear "backing."

A *feed-water heater* is located on the port side forward in engine room. It is composed of a composition cylindrical shell with flanges, to which are bolted at each end a tube-plate and a header. Expanded into the tube plates are one hundred thin, seamless-drawn ⅝-inch brass tubes, No. 14 B.W.G. thickness, 4 feet 3½ inches long. The heater is ar-

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ranged with feed-pipe connections, so that the discharge from the main-engine pumps and independent main-feed pump may be passed through or by-pass the heater. The auxiliary exhaust-steam pipe is connected with the heater shell near its upper end through a 4-inch nozzle, the outlet for the exhaust steam being at the side near the bottom. Three baffle plates through which the tubes pass, of rolled naval brass resting on four lugs each, guide the steam across and among the tubes. Suitable air cocks and drain pipe are fitted. The headers and tube sheets are stayed together by nineteen $\frac{5}{8}$-inch tobin-bronze socket bolts with brass pipe sleeves at each end. The feed water enters the bottom header, passes through the tubes and out at the top header. The headers are of composition with feed-pipe nozzles cast on. An adjustable automatic-spring relief valve on the auxiliary exhaust pipe at each main condenser regulates the exhaust pressure. *There are no auxiliary condensers.*

The *propellers* are of the three-bladed bent-back type, of manganese-bronze, true-screw, polished, tinned and repolished. The blades are cast with the hub. Two keys and nut on the taper end of the propeller shaft secure each propeller to its shaft. Over the nut is a composition cap or fairwater with watertight joint filled with tallow. These propellers were measured and fitted to guides or templets of thin wood, carefully arranged to conform to the drawings, and much time and care was devoted to making them true in every respect, including balancing. The following are the principal dimensions, etc., for one propeller.

PROPELLER DATA.

Number of blades	3
Diameter of propeller, feet and inches	7- 3
Pitch of propeller, feet and inches	6-10
Expanded area, square feet	15
Projected area, square feet	12.642
Ratio of pitch to diameter	1 : 1.06
Ratio of total expanded blade area to immersed midship section at trial draught	1 : 11.6
Ratio of expanded blade area to disc area	1 : 2.752
Angle of inclination of generatrix with axis of hub, degrees	77½

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The boilers are two in number, separated by an athwartship fireroom eleven feet wide, in one compartment. They are the Babcock & Wilcox Co.'s *Alert* type, and were constructed by that company at their works at Bayonne, N. J. As a complete description of this type of boiler has recently been published in the JOURNAL (Vol. XVI, p. 1223), a few particulars will suffice here, as follows:

FOR ONE BOILER.

Diameter of drum, inside, inches	42
Length of drum, neglecting heads, feet and inches	9- 6⅝
Thickness of drum plate, inch	21/32
Number of elements	15
Number of tubes, ordinary (2-inch)	108
upper, circulating (4-inch)	15
lower, circulating (4-inch)	2
side boxes, each side, rectangular	4
Length of tubes between headers, feet	9
over all, feet and inches	9- 2
Tube-heating surface, square feet	1,970.7
Plate-heating surface, side boxes and drum, square feet	108.8
Aggregate heating surface, square feet	2,079.5
Height of smoke pipe above grate, feet	60.33
Area through first section for gases of combustion, square feet	10.36
second section for gases of combustion, square feet	10.2
third section for gases of combustion, square feet	8.6
uptake section for gases of combustion, square feet	8.2
Sectional area of smoke pipe (37 inches diameter), square feet	7.4666
Ratio of heating to grate surface, fireside considered	41.68 : 1
Number of tubes, 2-inch, in intermediate sections	28
right-side section	23
left-side section	21
lower sections	2
4-inch tubes in each lower section	1
Thickness of header tube sheets, inch	½
Number of plates in shell of drum	1
Thickness of butt straps and tube-hole reinforcing, inch	21/32
Depth of header, outside, inches	5⅝
Width of header, outside, inches	6½
Weight of one boiler, complete, pounds	30,045

Boiler mountings include for each boiler a duplex 3-inch pop safety valve, two water gauges (one reflex and the other a tube protected by plate-glass shields), both in Dewrance fittings, with safety checks and quick-closing cocks by chains

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from fireroom; one main-feed stop and check valve, one auxiliary-feed stop and check valve; one main-steam stop valve with automatic steam closing appliance; one surface-blow valve and two bottom-blow valves and three gauge cocks. Each steam drum has near its highest part and at middle a fitting for two steam gauges, air valves and testing connection.

At the rear of each grate is a sprinkler for extinguishing fires in furnace, operated by a pipe conection and valve with the fire main at the back end of each boiler.

The ash pans are galvanized-steel and are shallow. The ash pits are fitted with automatically-closing doors which can be quickly lifted out and laid away if desired.

Forced Draft—Closed Fireroom System.—By closing trap doors over fireroom hatch on main deck, bulkhead doors between drum room and gun deck, and one between engine and firerooms, and starting the forced-draft fans over fireroom on gun deck level, the forced-draft system may quickly be put in operation and as quickly be discontinued. As the bunkers ventilate through permanently open ducts into the casing around smoke pipes, either the bunker doors should be kept closed, the exhaust nozzles in bunkers be covered, or a slight loss of efficiency will result. No gates are provided for the bunker-exhaust ducts. Two 40-inch centrifugal fan blowers, driven by 3×3-inch simple twin-cylinder engines with cranks opposite, are located on a level with the gun deck, over the fireroom, one forward on port side, the other aft on starboard side. They take air from two fireroom ventilators in their vicinity, which serve also as ash hoists.

The blower engines for forced draft are inverted direct-acting, mounted on forged-steel columns, have piston valves, are equipped with sight-feed and centrifugal oiling gear, and have long bearings. They perform excellently well on all occasions without heating of bearings at any speed. An indicator card from one of these engines was taken at 700 revolutions per minute, and at times their speed exceeded a thousand revolutions per minute without apparent injury or excessive wear.

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Number of blowers	2
Type of blower engine	Vertical, inverted, twin.
Diameter of cylinder, inches	3
Stroke, inches	3
Diameter of piston rod, inch	$\frac{5}{8}$
fan, inches	40
Width of fan at tips, inches	5
Area of induction nozzle, square feet	3.72 to 2.18
eduction nozzle (periphery of fan × width), square feet	4.35

Ash Hoists.—The fireroom ventilators are arranged with suitable guides and doors, as customary.

An engine of same form and size as the blower engine, *i. e.*, 3 × 3, is secured to the main-deck fireroom-hatch framing, amidships, and a spur gear on the engine shaft engages one larger on a drum shaft extending athwartships, abaft the engine and beyond both ventilators. A drum at each end of this shaft, mainly without, but whose periphery is within each ventilator, winds and unwinds the ash whips. A clutch on the drum shaft permits either drum to be used independently. A small hand lever starts the hoister, which latter stops automatically when the bucket reaches either end of the hoist. It was found desirable to introduce a reducing valve in the steam pipe to and near the ash-hoist engine, which provided safe working at the extremes of sea and port pressures, as the engine worked too violently under full-steaming pressures.

A bucket of ashes can be hoisted from fireroom to main deck in two seconds.

Trolley rails overhead on main deck, fitted with slides and hooks and unhooking levers, lead to ash chutes on each side of ship.

EVAPORATING AND DISTILLING PLANT.

Two evaporators of contractors' design and two distillers of Bureau of Steam Engineering design are installed; the former in the engine room, one each side, aft; the latter in the engine room hatchway, forward side. Each evaporator consists of a steel cylinder, with longitudinal axis vertical. Horizontal straight brass tubes expanded into headers are fitted into a composition rectangular frame in lower part of shell. Two

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steel diaphragms or baffles are in the steam space; the upper diaphragm, twelve inches from the top, is a perforated plate, ¼-inch holes; the lower is a conical plate, with a large central opening, hooded. The baffles are twelve inches apart. The inverted hood under the lower baffle is three inches below it, and dished on its edges downward. The upper head is inserted, and is dished to a radius of about five feet. The bottom is flat, inserted, stiffened by two angles, and riveted to shell with flange external. Each header has two vertical stiffening ribs and stay bolts.

DATA FOR EACH EVAPORATOR.

Diameter shell, inside, inches	33
Length over all, feet and inches	5- 9
Thickness of shell, inch	$\frac{3}{16}$
heads, inch	$\frac{5}{8}$
Number of tubes	72
Diameter of tubes, inches (outside diameter)	2
Thickness of tubes, No. B. W. G	12
Length of tubes, inches	$28\frac{5}{8}$
between tube plates, inches	$26\frac{3}{4}$
Pitch of tubes, inches	$2\frac{3}{4}$
Thickness of tube plates, inch	$\frac{7}{8}$
header plates, inch	1
Number of stay bolts in each header	16
Diameter of stay bolts in each header, inch	$\frac{7}{8}$
Material of headers	Composition
stay bolts	Tobin bronze.
Heating surface, square feet	84.04
Capacity of evaporators combined, nominal, gallons	3,500
Actual output on test, rate per diem, gallons	4,034

Distillers.—Two of Bureau design, each with 32 square feet cooling surface, are located side by side on forward bulkhead of engine-room hatch between main and gun-deck gratings. As this type of distiller is so well known in the service, a slight description only should be essayed: Each consists of a vertical cast-iron cylinder, with flanges for bolting tube plate and header together at top end and a stuffing box and packing gland and bolts at the bottom end. Straight tubes are expanded at one end into a composition tube sheet, which is secured between the upper flange of the cylinder shell and the aforesaid header. The lower ends of the tubes are expanded

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into a cylindrical box or header, which latter is free to move by the expansion of the tubes in the stuffing box.

The vapor from the evaporators is admitted through a nozzle close to the top at side of shell, around (*i. e.*, outside) the tubes, while the circulating water is introduced at the bottom header, and, passing through the tubes, makes its exit, at the top of the top header.

At the vapor nozzle within the distiller shell is a perforated galvanized sheet-iron baffle ring surrounding the tubes at the upper end, with an axial length of $3\frac{1}{8}$ inches and inside diameter of $7\frac{3}{4}$ inches. This baffle ring is flanged at both ends to $8\frac{15}{16}$ inches diameter, which is nearly the inside diameter of the distiller shell. By the upper flange the baffle ring is secured by four small brass screws to the upper tube plate. The perforations in the baffle plate are 64 of $\frac{1}{4}$-inch diameter.

The distillate is drawn off at a nozzle at the lower side of shell, and flows by gravity or may be pumped to the ship's tanks or through the fresh-water system. Intermediate between the distillers and fresh-water pump adjacent is a small galvanized-iron reservoir of about three gallons capacity, with removable cover, which can be filled and emptied through the fresh-water pump suction from distillers. A water meter is on the same line of pipe between distillers and said reservoir.

DATA FOR EACH DISTILLER.

Diameter, shell, outside, inch	$9\frac{3}{8}$
inside, inches	9
Length, shell, between flanges, outside, feet and inches	4– $3\frac{3}{8}$
distiller over headers, feet and inches	4–$10\frac{3}{8}$
Thickness, upper tube sheet, inch	$\frac{3}{4}$
lower tube sheet, or header sheet, inch	$\frac{3}{8}$
Number of tubes	46
Diameter of tubes, outside, inch	$\frac{5}{8}$
Length of tubes (over all), feet and inches	4– $1\frac{3}{8}$
between headers, feet	4
Cooling surface, outside, square feet	nearly 30.2

OILING ARRANGEMENTS.

Galvanized steel oil tanks for storage of lubricating, cylinder and lamp oil are located in the after end of the engine room, secured to athwartship bulkhead, one for each kind of oil, the form and capacity of which are, for the lubricating

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oil: rectangular, 66 × 18 × 60 inches inside measurements, capacity 300 gallons; for the cylinder and lamp oils, one each rectangular, 18 × 18 × 36 inches, of 50 gallons capacity each.

The above-named tanks are filled through pipe from gun deck. A gravity system of filling the oil cups on the engine is supplied from a reservoir, cylindrical tank, of about 10 gallons capacity, secured between the distillers and feed-water pump in the engine-room hatch. A hand pump near by with piping from main oil tanks is fitted for filling this distributing tank.

Centrifugal oilers are attached to each crank pin of main and auxiliary machinery. Sight-feed oil boxes of polished cast brass are fitted each with twelve adjustable oil-feed valves and tubes for wicks, so that either or both wicks and valves may be used.

Each box contains swash plates, and each drip tube leads through a vertical glass tube, suitably dust proof, to the copper pipes to the different bearings.

Each main engine is fitted with four of these sight-feed distributing boxes. These boxes are supplied from a ½-inch brass pipe leading from the gravity tank fore and aft over the engine. This oil-distributing main is connected by a vertical pipe with a horizontal ⅜-inch brass pipe attached to the outboard side of condenser, supplying the oil cups on main crank-shaft bearings, of which there are six, to each engine.

The eccentric straps and similar oscillating gear are oiled through long, rectangular, open-top cups filled with curled hair, the supply of oil coming from the aforesaid distributing boxes about the top of the cylinders.

COOLING-WATER SERVICE.

A complete cooling-water service for all engine bearings, including line, shaft and thrust bearings, is installed in the usual manner. The cooling-water main supply is taken from the condenser-circulating outlet chamber, from the intake to circulating end of condenser, or may be forced by a hose connection from the fire pump in engine room. Spraying nozzles

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PUMPS AND CONNECTIONS.

No.	Pumps	Size (inches).	Kind.	Suction pipes from—		Discharge pipes to—		Location.
				Inches.		*Inches.*		
2	Main air	12 dia.; 6.9 stroke	Worked from I.P. crosshead	4	Condenser	3	Main feed tanks	1 on each engine.
2	Main circulating	Centrifugal	Vertical compound, 18-in. runner	5½	Sea	5½	Cond's'r and overboard.	1 for each condenser.
				5½	Main drain.			Each side E. R.
				5½	Bilge.			
2	Hotwell	4½ x 6 x 6, 95 gals	Vertical piston, double-acting, single, light service.	2½	Either air-pump suction	2½	Main feed tanks	In engine room.
				2½	C. & R. pipes from ship's sides.	2½	Reserve feed-water tanks.	Forward bulkhead between feed tanks.
				2½	Reserve feed-water tank.			
2	Main feed (attached)		Worked from I.P. crosshead	2	Feed-tank connecting pipe.	2	Main feed discharge main.	1 on each engine.
1	Main feed (independent).	9 x 5¼ x 8, 90 gals	Piston plunger, double-acting, single	2½	Feed-tank connecting pipe.	2½	Main feed discharge main.	In engine room, port side.
1	Auxiliary feed (independent).	9 x 5¼ x 8, 90 gals	Piston plunger, double-acting, single.	2½	Feed-tank connecting pipe.	2½	Auxiliary feed main.	In boiler room.
				2½	Auxiliary drain	2½	Fire main.	
				2½	Fire room bilge	2½	Overboard.	
				2½	Sea.			
				2½	Bottom blows	2½	Hose connection.	
				2½	C. & R. suction.			
				1½	Hose connection.			
1	Fire and bilge	8 x 6 x 12, 147 gals	Vertical piston, double-acting, single.	3	Sea	3	Fire main	Engine room.
				3	Main drain	3	Overboard.	
				3	Auxiliary drain	2½	Hose connection.	
				3	Drainage manifolds.			
				3	Engine-room bilge.			
				3	Crank pits.			
				1½	Hose connection.			
1	Bilge	3½ and 5½ diam.; 4.95 stroke.	Worked from I.P. crosshead	3	Engine-room bilge	3	Overboard	On starboard engine.
				3	Crank pits, each engine.			
1	Sanitary	3½ and 5½ diam.; 4.95 stroke.	Worked from I.P. crosshead	3	Sea	3	Flushing system	On port engine,
1	Distiller circulating	6 x 8 x 12, 261 gals	Vertical piston, double-acting, single, light service.	4	Sea	4	Distiller	Where directed.
						2½	Flushing system	
1	Evaporator feed	3½ by 2½ by 4	Light service		Sea		Evaporators	After bulkhead.
					Distiller, circulating water discharge.			
1	Distiller fresh water	Of approved capacity; 3 x 2¼ x 4.	Vertical piston, light service		Distillers, gravity discharge.		Fresh water tanks. Main-feed tanks. Distiller gravity discharge.	E. R. hatch.
1	Brine	3 x 2¼ x 4	Light service		Bottom of evaporators		Overboard	After bulkhead.

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are fitted over each crank-shaft bearing, crossheads and eccentrics.

ENGINE TRAMS.

To locate the proper (original) position of the main valve when the engine under consideration is "on the center," trams are made, two for both engines; also a tram for placing any crank "on the center," and bridge trams for locating the true (original) position of the crank-shaft journals in reference to their pillow blocks were fitted and neatly stowed in boxes for safe keeping. For the *Dubuque* and *Paducah* these latter trams proved to be exactly alike in all dimensions, so that one might be used for either ship, indicating a close or similar adherence to the drawings in both cases, in this particular at least. Generally speaking, the workmanship on these engines is excellent.

THE TRIAL OF U. S. S. *DUBUQUE*.

On the 5th day of May, 1905, the *Dubuque* was steamed from the contractors' works to the Navy Yard, New York, where the next day the ship was coaled with government coal for the purpose of bringing the ship down to the normal draught. Contractors' coal for the trial had been previously placed on board, weighed in bags, for convenience in tallying during the trial. This latter coal was Pocahontas hand-picked, and of excellent quality. It was stowed on the gun deck in temporary bins constructed for the purpose, and during the trial was taken into the fireroom through the bunkers, in bags, as wanted. On the 7th of May the vessel steamed to Newport, R. I., through the Sound, making the distance in eleven hours against a tide nearly all the way. The machinery on this run performed perfectly. On this run indicator cards were taken from the auxiliary machinery.

After taking aboard water in both trimming tanks, reserve water tanks and in water bottoms, to bring the vessel to a mean draught of 12 feet 3 inches, the progressive speed runs were made over a measured mile in Narragansett Bay on the afternoon of the 10th. A record of these runs is shown in the table and by the speed curves which follow.

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Table No. 1.—U. S. S. *DUBUQUE*.

Data from progressive runs over measured-mile course off Newport, R. I., May 10, 1905.

Number and direction of run.	Time on course.	Revolutions of main engines. To make 1 knot. Starboard.	Revolutions of main engines. To make 1 knot. Port.	Revolutions of main engines. Per minute. Starboard.	Revolutions of main engines. Per minute. Port.	Revolutions of main engines. Per minute. Mean.	Speed in knots per hour.	Indicated horsepower. Starboard engine. H. P. cylinder.	Indicated horsepower. Starboard engine. I. P. cylinder.	Indicated horsepower. Starboard engine. L. P. cylinder.	Indicated horsepower. Starboard engine. Total.	Indicated horsepower. Port engine. H. P. cylinder.	Indicated horsepower. Port engine. I. P. cylinder.	Indicated horsepower. Port engine. L. P. cylinder.	Indicated horsepower. Port engine. Total.	Indicated horsepower. Total both main engines.	Indicated horsepower. Auxiliaries in operation (estimated).	Indicated horsepower. Total all machinery.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
1-S.	6.6233	794.9	882.4	120.1	133.2	126.6	9.06	24.34	29.91	17.82	72.07	28.08	40.82	29.47	98.37	170.44	26.67	197.11
2-N.	8.5233	1,176.2	1,124.6	138	131.9	135	7.04	37.89	46.60	23.83	108.34	27.59	32.84	17.79	78.22	186.56	25.38	211.94
3-S.	5.7333	883.5	894.5	154.1	156	155.1	10.464	54.39	60.51	33.26	148.16	53.02	53.79	39.56	146.37	294.53	25.47	320
4-N.	6.6733	1,095.2	1,113.1	164.1	166.8	165.5	8.991	64.36	76.63	52.24	193.23	66.24	71.89	54	192.13	385.36	27.08	412.44
5-S.	5.3167	963.5	935.4	181.2	175.9	178.6	11.284	88.24	109.4	83.1	280.74	73.01	78.34	61.69	213.04	493.78	29.5	523.28
6-N.	5.49	1,069.4	1,111	194.8	202.4	198.6	10.93	91.68	128.7	98.8	319.18	110.14	121.16	109.2	340.5	659.68	34.86	694.54
7-S.	4.74	1,000.1	1,024.5	211	216.2	213.6	12.658	127.6	169.8	148	445.4	135.67	163.58	154.57	453.82	899.22	38.8	938.02
8-N.	4.8367	1,091.7	1,102	225.7	227.8	226.8	12.405	143.8	207.5	199.7	551	158.58	181.08	180.68	520.34	1,071.34	45.8	1,117.14
9-S.	4.4717	1,055.9	1,079	236.1	241.3	238.7	13.417	181.3	248.7	262.4	692.4	203.08	231.11	252.58	686.77	1,379.17	75.06	1,454.23
10-N.	4.53	1,083.4	1,107.4	239.4	244.5	241.8	13.245	164.3	256.8	271.3	692.4	197.38	240.02	260.54	697.94	1,390.34	56.11	1,446.45
11-S.	4.5133	1,068.2	1,091.8	236.7	242	239.3	13.294	154.7	259.6	258.6	672.9	198.92	229.46	246.78	675.16	1,348.06	58.86	1,406.92

Table 1-A.—TRIALS U. S. S. *DUBUQUE*.—HORSEPOWER OF AUXILIARIES.

No. of runs.	Circulating pump.	Main feed.	Auxiliary feed.	Bilge.	Water service.	Blowers (2).	Dynamos.	Steering engine.
1	S. 0.39, P. 0.48	5	1.5	.....	0.5 (estimated).	S. 1.4, P. 1.4	15	1
2	S. 0.39, P. 0.49	4.2	1	.....	0.5 (estimated).	S. 1.4, P. 1.4	15	1
3	S. 0.48, P. 0.49	4.4	1.2	.....	0.5 (estimated).	S. 1.3, P. 1.1	15	1
4	S. 0.5, P. 0.48	4.8	1.4	1 (estimated).	0.5 (estimated).	S. 1.3, P. 1.1	15	1
5	S. 0.5, P. 0.4	4.8	1.3	1 (estimated).	0.5 (estimated).	S. 3.2, P. 2.5	15	1
6	S. 0.37, P. 0.39	5	1.3	1 (estimated).	0.5 (estimated).	S. 5.1, P. 5.2	15	1
7	S. 0.7, P. 0.4	7.2	1.3	1 (estimated).	0.5 (estimated).	S. 5.7, P. 6	15	1
8	S. 0.73, P. 0.77	13.3	1.4	1 (estimated).	0.5 (estimated).	S. 6.1, P. 6	15	1
9	S. 0.78, P. 0.78	17	1.6	1 (estimated).	0.5 (estimated).	S. 11.5, P. 11.5	15	1
10	S. 1.02, P. 1.09	13.1	2	1 (estimated).	0.5 (estimated).	S. 10.7, P. 10.7	15	1
11	S. 1.03, P. 0.83	16	2.1	1 (estimated).	0.5 (estimated).	S. 10.7, P. 10.7	15	1

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Table No. II.—U. S. S. *DUBUQUE*.

Data from Progressive Runs over Measured-Mile Course off Newport, R. I., May 10, 1905.

Number of run.	Time at beginning of run P. M. (approximate).	Which engine.	Steam pressures. In boilers above atmosphere.	Steam pressures. At engine above atmosphere.	Steam pressures. H.P. steam chest, absolute.	Steam pressures. In first receiver, absolute.	Steam pressures. In second receiver, absolute.	Mean effective pressures. H.P. cylinder.	Mean effective pressures. I.P. cylinder.	Mean effective pressures. L.P. cylinder.	Mean effective pressures. Referred to low pressure.	Vacuum, inches.	Cut-off in cylinders from beginning. H.P.	Cut-off in cylinders from beginning. I.P.	Cut-off in cylinders from beginning. L.P.	Temperatures. Outside air.	Temperatures. Engine-room working platform.	Temperatures. Fireroom working level.	Temperatures. Injection.	Temperatures. Discharge.	Temperatures. Hotwell tank.	Temperatures. Feed.	Revolutions or double strokes. Circulating pump.	Revolutions or double strokes. Main-feed pump.	Revolutions or double strokes. Auxiliary-feed pumps.	Revolutions or double strokes. Air-pump discharge temperature.	Revolutions or double strokes. Water-service pump.	Revolutions or double strokes. Bilge pump.	Revolutions or double strokes. Fire-room blowers.	Air pressure, inches water.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
	h m																													
1	2-22	St'b'd	170	178	85	20	8	31	13	2.75	11.13	25	H.P., 0.751.	I.P., 0.723.	L.P., 0.695.	Outside air, 75.	85	108	50	66	83	162	210	30	9	76	18	...	350	...
		Port	178	155	80	24	14	32.25	16	4.1	13.68	28					82		45	76	75		220						350	
2	2-45	St'b'd	165	178	89	27	9	42	17.63	3.2	14.55	25.5					87	119	50	68	93	180	210	24	5	75	17	...	350	...
		Port	165	180	75	20	9	32	13	2.5	10.99	28					76		45	69	97		225			79			350	
3	3-12	St'b'd	168	168	120	37	10	54	20.5	4	17.82	25.7					87	119	50	70	89	207	224	26	6	70	17	...	325	...
		Port	165	173	105	30	9	52	18	4.7	17.39	28.5					76		48	76	88		225			90			300	
4	3-30	St'b'd	178	168	135	43	13.5	60	24.38	5.9	21.83	26 2					87	117	49	74	84	190	220	28	8	70	12	50	325	...
		Port	175	174	125	35	11	60.75	22.5	6	21.35	28.5					76		50	81	83		220			84			300	
5	3-47	St'b'd	165	163	165	61	23	74.5	31.5	8.5	28 71	26.5					87	118	49	83	88	179	230	28	7	72	12	50	600	.2
		Port	160	170	134	40	13	63.5	23.25	6.5	22.45	28					77		50	80	84		215			84			550	
6	4- 4	St'b'd	180	188	185	70	25	72	34.5	9.4	30.37	26.5					87	111	48	90	92	174	205	30	7	78	14	50	825	.45
		Port	177	165	170	57	18	83.25	31.25	10	31.18	27.5					78		50	94	85		210			94			830	
7	4-19	St'b'd	170	233	235	90	28	92.5	42	13	39.12	26.5					87	112	49	86	86	177	270	38	7	98	17	50	900	.2
		Port	167	214.5	210	75	24	96	39.5	13.25	38.9	27.5					79		50	94	98		215			96			925	
8	4-33	St'b'd	245	248	265	95	30	97.5	48	16.4	45.25	26.5					87	112	49	95	86	175.5	275	50	8	100	20	55	950	.3
		Port	240	228	220	80	27	106.5	41 5	14.7	42 33	28					80		50	82	104		300						940	
9	4-52	St'b'd	275	248	265	105	34	117.5	55	20.6	54.36	26.5					87	116	48	100	102	183.5	305	68	60	105	20	55	1,600	.6
		Port	275	248	265	98	34	128 75	50	19 4	52.75	27.5					81		50	106	106		305						1,600	
10	5- 5	St'b'd	275	268	270	108	34	105	56	21	53.6	26 5					87	114	49	99	120	179	342	49	11	100	21	50	1,500	.45
		Port	271	258	265	100	34	123.5	51.25	19.75	52.9	27.5					81		50	90	104		350						1,500	
11	5-18	St'b'd	273	268	270	105	34	100	57.25	20.25	52.7	26.5					87	118	49	105	120	181	344	54	12	136	20	55	1,500	.5
		Port	275	258	265	100	34	125.75	49.5	18.9	51.7	26					81		50	122	117		310						1,500	

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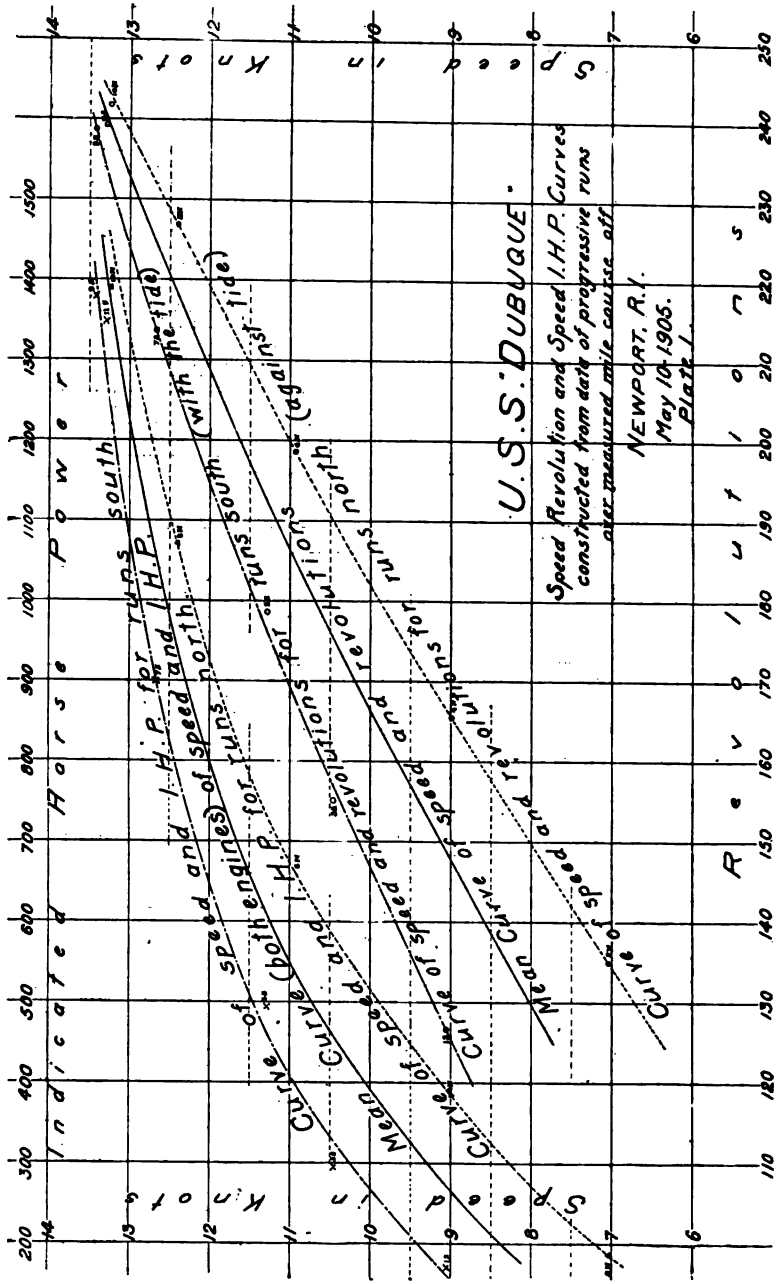
U.S.S. "DUBUQUE"
Speed Revolution and Speed I.H.P. Curves constructed from data of progressive runs over measured mile course off NEWPORT, R.I.
May 10 1905.
Plate I.
Indicated Horse Power
Revolutions
Knots
Speed
I.H.P. for runs south
I.H.P. for runs north
Curve of speed and I.H.P. (both engines) of speed and north
Curve of speed and revolutions for runs south (with the tide)
Curve of speed and revolutions for runs north (against tide)
Mean Curve of speed and revolutions
Curve of speed and I.H.P.

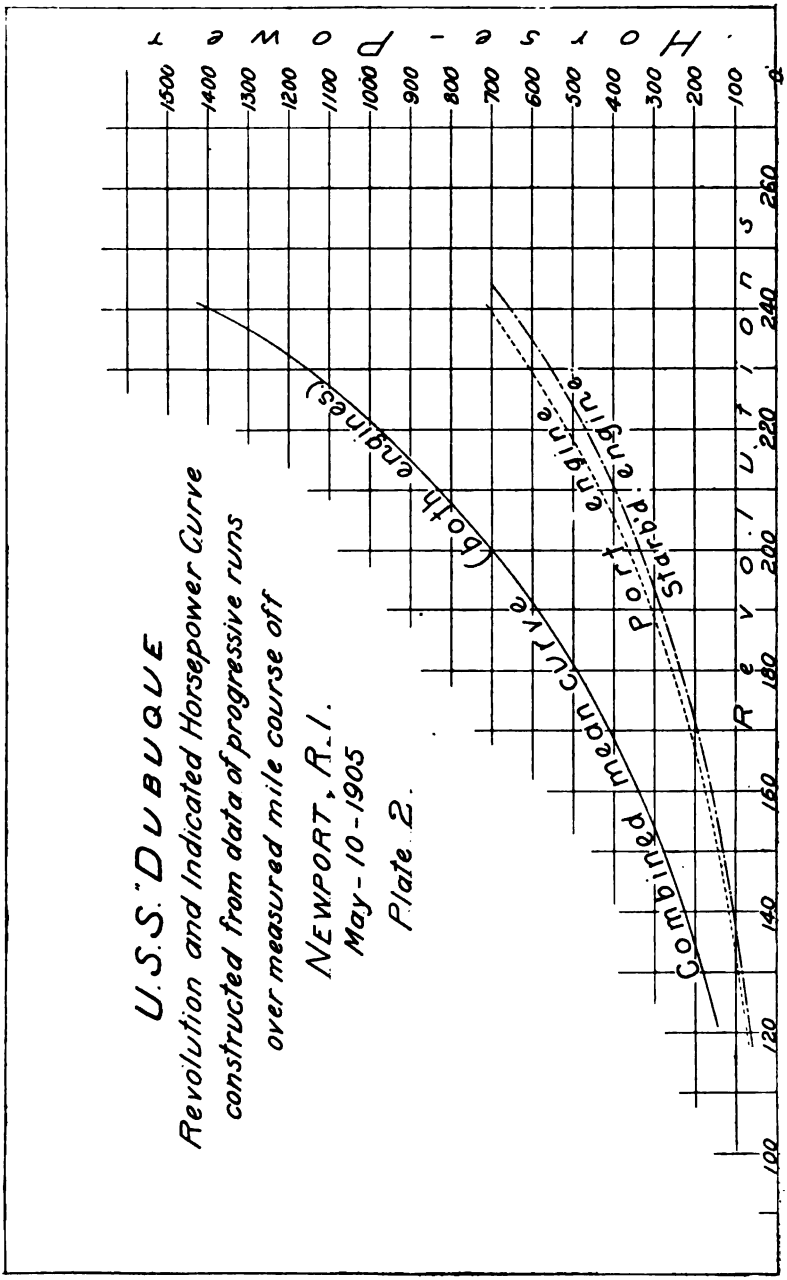

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Table III.—U. S. S. *DUBUQUE.*

Data Deduced from Progressive Runs over Measured-Mile Course off Newport, R. I., May 10, 1905.

Speed in knots per hour.	Revolutions per minute.	Indicated horsepower.			Slip of propeller in per cent. of its own speed.	Constant R + S.
		Starboard engine.	Port engine.	Total both engines.		
7	112.5	52	60	112	7.72	16.000
7.1	114.2	56	62	118	7.81	16.085
7.2	116	58	66	124	7.95	16.110
7.3	117.6	60	68	128	7.94	16.110
7.4	119.3	64	72	136	8.01	16.120
7.5	121	68	76	144	8.08	16.135
7.6	122.7	72	80	152	8.15	16.140
7.7	124.4	73	83	156	8.21	16.155
7.8	126.2	76	90	166	8.35	16.180
7.9	128	80	92	172	8.47	16.203
8	129.8	85	95	180	8.59	16.22
8.1	131.5	89	97	186	8.66	16.230
8.2	133.2	94	102	196	8.72	16.245
8.3	135	96	108	204	8.82	16.265
8.4	136.8	100	110	210	8.94	16.280
8.5	138.6	105	115	220	9.06	16.305
8.6	140.4	109	119	228	9.25	16.325
8.7	142.2	111	125	236	9.27	16.343
8.8	144.1	118	128	246	9.43	16.373
8.9	146	122	132	254	9.60	16.400
9	147.7	127	137	264	9.64	16.410
9.1	149.7	130	144	274	9.85	16.450
9.2	151.4	135	149	284	9.88	16.456
9.3	153.3	142	154	296	10.03	16.480
9.4	155.2	146	162	308	10.18	16.510
9.5	157	151	167	318	10.27	16.525
9.6	158.8	156	172	328	10.34	16.540
9.7	160.7	164	178	342	10.48	16.565
9.8	162.6	169	185	354	10.61	16.590
9.9	164.4	175	193	368	10.70	16.605
10	166.4	185	199	384	10.89	16.640
10.1	168.2	188	206	394	10.96	16.650
10.2	170.2	196	216	412	11.13	16.685
10.3	172.1	204	222	426	11.26	16.710
10.4	174.1	210	230	440	11.42	16.740
10.5	176.1	220	240	460	11.59	16.770
10.6	178.1	229	245	474	11.75	16.802
10.7	180.1	236	256	492	11.90	16.830
10.8	182.2	246	264	510	12.11	16.870
10.9	184.4	255	277	532	12.35	16.920
11	186.3	264	284	548	12.44	16.935
11.1	188.4	270	300	570	12.64	16.970
11.2	190.5	285	309	594	12.82	17.100
11.3	192.8	293	322	615	13.10	17.060
11.4	195	307	334	641	13.31	17.105
11.5	196.1	314	340	654	13.03	17.050
11.6	199.4	330	360	690	13.74	17.190
11.7	202	347	377	724	14.11	17.265
11.8	204	360	390	750	14.22	17.290
11.9	206.2	372	408	780	14.42	17.330
12	208.6	388	424	812	14.70	17.380
12.1	211	402	442	844	14.96	17.435
12.2	213.5	420	460	880	15.27	17.500
12.3	215.7	434	478	912	15.44	17.540
12.4	218.1	450	500	950	15.70	17.590
12.5	220.5	469	519	988	15.94	17.640
12.6	223	490	540	1,030	16.21	17.700
12.7	225.2	510	560	1,070	16.38	17.730
12.8	227.7	530	582	1,112	16.64	17.790
12.9	230.2	552	607	1,162	16.90	17.850
13	232.6	576	630	1,206	17.09	17.890
13.1	235.2	608	656	1,264	17.40	17.955
13.2	237.8	634	686	1,320	17.69	18.015
13.3	240.5	664	712	1,376	17.99	18.082

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SMOKE OBSERVATIONS MADE DURING THE THREE MAXIMUM SPEED RUNS OF THE PROGRESSIVE-SPEED TRIAL.

Time smoke appeared.			Time smoke disappeared.			Duration of smoke.
Hrs.	*Min.*	*Sec.*	*Hrs.*	*Min.*	*Sec.*	*Sec.*
4	48	35	4	49	20	45
	50	35		51	30	55
	52	35		53	30	55
	54	30		55	20	50
	56	43		57	25	42
	58	43		59	30	47
5	00	54	5	1	50	56
	2	40		3	20	40
						Average, 48

One raked and one door fired every two minutes.

THE FULL-SPEED FOUR-HOUR (PRELIMINARY) TRIAL.

On the 11th of May, at 8.45 A. M., at sea, the contract full-speed trial was begun and proceeded without mishap of any kind. For the purpose of avoiding a long and useless return run in addition to the trial run, the latter included a very gradual turn so as to bring the finish somewhere near the starting point.

The performance was in every respect satisfactory to the trial board. The ship steered well, turned in a small circle and was free from unusual vibrations. The vibrations of the main engines, never excessive, were greatest at 120 revolutions; at 220 revolutions it was almost *nil.*

All bearings ran cool; no water being used except on the thrust bearings as a precaution. Indicator cards were taken from all main cylinders every half hour simultaneously, and all engine data was recorded every fifteen minutes. Cards from the auxiliaries had been taken at various speeds previously.

The firing or stoking was systematically performed under the direction of a representative of the Babcock & Wilcox Co., at regular intervals of two minutes, thus:

Time.	Rake.	Fire.
4·36	2	1
38	3	2
40	1	3
42	2	1
44	3	2
46	1	3
48	2	1
50	3	2

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The numbers indicate the furnace doors of each boiler; three furnace doors, one furnace to each boiler. Thus the fires of one-third of one furnace were being raked while fires of an equal portion of the other boiler were being coaled. A certain number of shovelfuls were thrown on at each firing, no hesitation or study on the part of the fireman being necessary. Considering the boilers, the method of firing and general excellence of the machinery, it is not surprising that the results were, for naval engines, so satisfactory.

As so much has been written about the smoke nuisance of certain of our ships, and for purposes of comparison, photographs were taken at regular intervals to include all conditions of firing during the four-hour run, showing the tops of the funnels and smoke issuing, which, by the way, was very slight. There was no smoke trail whatever.

PERFORMANCE ON PRELIMINARY TRIAL, MAY 11, 1905.

Trial began	8·45 A. M., May 11, 1905
Duration of trial (two hours seaward, a long turn and return), hours	4
Steam pressure at boilers (per gauge), pounds	240.8
H.P. steam chest, S. engine (per gauge), pounds	222.7
P. engine (per gauge), pounds	221.2
1st receiver, S. engine (absolute), pounds	94.2
P. engine (absolute), pounds	87
2d receiver, S. engine (absolute), pounds	29.4
P. engine (absolute), pounds	30

MEAN EFFECTIVE PRESSURES IN CYLINDERS IN POUNDS PER SQUARE INCH.

Main engines:	
Starboard	H.P., 105.38; I.P., 51.39; L.P., 17.39
Port	H.P., 113.87; I.P., 45.10; L.P., 17.01
Mean pressure in pounds per square inch on L.P. piston equivalent to aggregate M.E.P. on all pistons, starboard	48.41
Mean pressure in pounds per square inch on L.P. piston, equivalent to aggregate M.E.P. on all pistons, port	46.82

REVOLUTIONS PER MINUTE.

Main engines	starboard 227.73; port 231.79
Pumps, air	starboard, attached; port, attached.
Pumps, circulating	starboard 264.6; port 311
feed, double strokes per minute	46.5
auxiliary, double strokes per minute, water service	14
bilge	50

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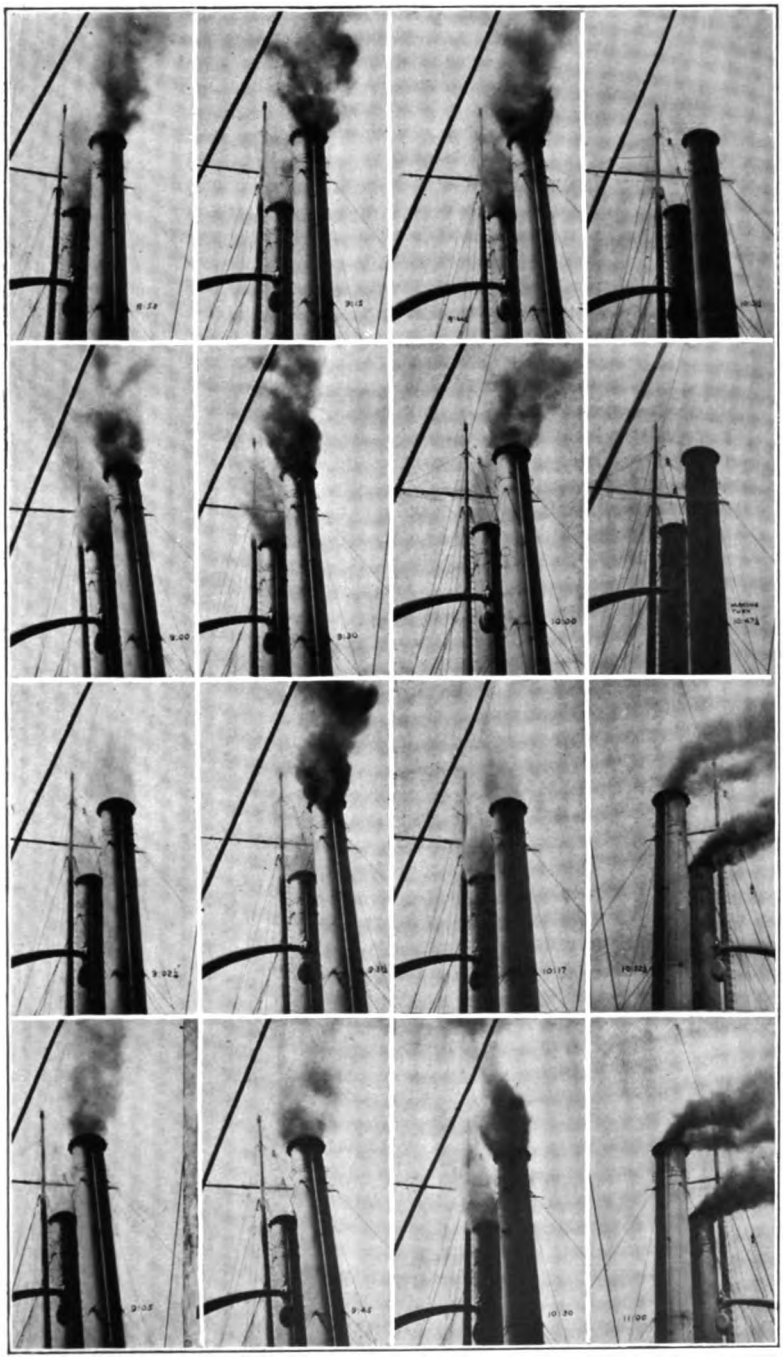

U. S. S. "DUBUQUE."

Photographs of Smoke taken during Official Trial.

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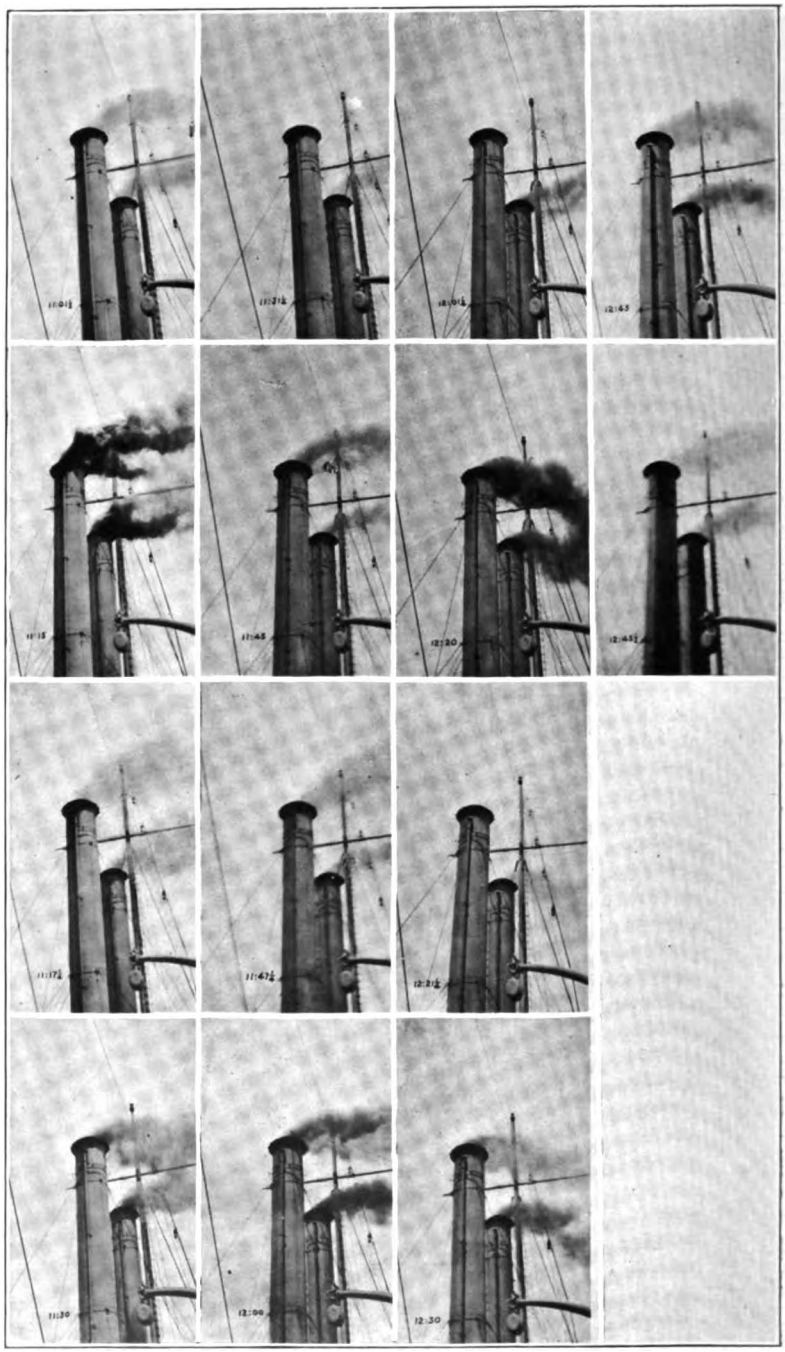

U. S. S. "DUBUQUE."

Photographs of Smoke taken during Official Trial.

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Blower engines.....forward, 447.7; after, 451.1
Speed of ship, in knots per hour..... 12.90
Slip of propeller, in per cent. of its own speed— { Based on mean pitch. St'b'd... 16.32 Port... 17.79 Based on final pitch. St'b'd... 16.32 Port... 17.79 }
Air pressure in firerooms, in inches of water..... .125
at base of stack, in inches of water..... .5
Temperature at base of stack, degrees Fahrenheit..... 569.4
outside air, degrees Fahrenheit..... 77
fireroom, working level, degrees Fahrenheit..... 112

INDICATED HORSEPOWER.

Main engines:
Starboard.....H.P., 156.86; I.P., 224.17; L.P., 213.67; Total, 594.70
Port.....H.P., 172.52; I.P., 200.24; L.P., 212.73; Total, 585.49
Air pump, starboard engine.....Attached. Port engine.....Attached.
Circulating pump, starboard engine.. .62 Port engine..... .85
Feed pumps, main, one only installed..... 11.3
(Attached pump not used on trial.)
Bilge pump..... 1
Water-service pumps on sanitary service..... .5
Ventilating engines for ship, electric; for engine rooms, electric.
Blower engines for forced draft in firerooms (2), total..... 4.2
Dynamo engines..... 21.4
Collective of all main engines..... 1,180.19
main engines, air, circulating and feed pumps..... 1,192.96
and auxiliary engines in operation during trial... 1,220.06
all machinery during trial, per square foot of G.S..... 12.228
H.S..... .293
Main engines, air, circulating and feed pumps, per square foot of G.S..... 11.95
Main engines, air, circulating and feed pumps, per square foot of H.S..... .287

COAL.

Kind and quality.....Pocahontas hand-picked, excellent.
Pounds per hour..... 1,903.52
Pounds per hour, per I.H.P., collective, all machinery in operation..... 1.559
Pounds per hour, I.H.P., collective, main engines, air, circulating and feed pumps..... 1.596
Pounds per hour, per square foot of G.S..... 19.08
H.S..... .4577
Cooling surface, square feet per I.H.P..... 1.113
Heating surface, square feet per I.H.P..... 3.409

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THE SMOKE QUESTION.

RELATION BETWEEN SMOKE DEVELOPMENT AND UTILIZATION OF FUELS, AND THE WAYS AND MEANS OF SMOKE ABATEMENT DURING FIRING.

By F. Haier, Hamburg.*

Translated from the "Zeitschrift des Vereines Deutscher Ingenieur" by William Wachsmann, Associate Member.

Gentlemen, respecting my presentation I must remark that, owing to the extent of the field in question and the difficulties met with, a complete and exhaustive treatment cannot be considered. In the following I will restrict myself principally to the field of steam-boiler furnaces, and of this only so much shall be considered as will be necessary to indicate the way in which, in my opinion, a healthy progress must be directed.

In the course of this discussion I will show how the *Society for the management of furnaces and controlling of smoke*,† formed in Hamburg in 1902, took hold of this problem, and how improvement in this field is to be brought about.

As to the annoyances, burdens and damages produced by smoke, further comment is unnecessary. We all concede that smoke is no agreeable accompaniment to firing. But, on the other hand, this much is also established, that as long as we

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* Lecture delivered before the Hamburg and Berlin branches of the Society of German Engineers.

† The circumstances surrounding the formation of this Society were brought about by the contemplated action of the Hamburg municipal authorities to proceed against the smoke and soot evil of industrial furnaces. Against the issuing of such rules, however, vigorous opinions became manifest in manufacturing circles. The committee on industry of the Chamber of Commerce expressed themselves in the senate to the effect that improvement of the conditions in question could be more certain and more easily obtained by way of self help—the industrial circles interested coming together and forming a society along the lines of the Boiler Guardian Society, and declaring that they were ready to form such a society. Thereupon the contemplated municipal measures were dropped to await the activity of this association, which, in the fall of 1902, after the completion of certain preliminaries, took up its labors, and for its object had "the procuring of the most perfect and smokeless combustion with the greatest possible utilization of fuel in the installations of its members." According to the by-laws of the Society this object was particluarly to be brought about by:

"1. Regular inspection of boiler plants, including the supervision of their service and proposals to remedy any defects that might be found.

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are compelled to burn coal complete prevention of smoke is out of the question. We must rather be satisfied to restrict this evil within certain practically-attainable limits. Furthermore, while industrial furnaces are generally held responsible for these annoyances, there is no doubt that in many cities during the winter months the domestic furnaces are no less the transgressors; but to get at these is more difficult than in the case of industrial furnaces.

The difficulties of the question can be seen from the attempts made for relief by means of laws. In no instance up to the present time have such laws produced any satisfactory results. Recalling the case of England, for example, where this question, due to the early development of its industries, first came up, no considerable progress has as yet been made despite the existence, for decades, of a smoke-prohibition law. Even with us, in Germany, attempts to progress by way of municipal measures have also been made at different places. If success has not attended these efforts to adjust the conflicting interests, then, in my opinion, it is because, in the case of this complicated problem, many facts remain to be established.

There is hardly any branch of engineering in which so many contradictory views are still held and, in spite of the extraordinary inventive ability displayed, no such extreme methods of practice are to be found as in this field; this more than anything else, it seems, would show that much which is essential yet remains to be made clear.

Before entering upon the discussion of the ways and means

"2. Dealing with all questions, such as the management of furnaces, utilization of fuels, smoke production, the plant and the modification of furnaces, heat value of fuels and questions relating thereto, the giving of opinions concerning all these, also the supervision of plants in the course of construction and those to be rebuilt, etc.

"3. The education of firemen.

"4. The collection of information obtained from the actual operation of such plants, particularly the testing of special firing contrivances, and the dissemination of the same among its members."

The members who, besides the payment of annual dues, pay a yearly fee for each furnace plant and fees for special work (evaporating tests, opinions, consultations), may demand that their furnace plants be regularly examined at least three times a year. They pledge themselves to follow the instructions of the Society's officers and to notify its technical staff of any new installations or rebuilding of existing ones which may be contemplated.

For the development of the Society and its method of procedure see its first annual report, 1902 to 1903 (published by Boysen & Maasch, Hamburg).

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of smoke abatement, there is one point I would like to examine more closely which, above all, is of vital interest to industry and is also of fundamental importance in judging the whole question, viz: the relation between smoke development and the utilization of fuels. Upon this question the views held in engineering circles are still far apart and the clearing up of this in the beginning would, in itself, be an appropriate contribution toward a healthy progress.

I. RELATION BETWEEN SMOKE DEVELOPMENT AND THE UTILIZATION OF FUELS.

In firing coal rich in gas, therefore inclined to make smoke, good heat utilization is not always combined with light smoke or smokeless working. Both *might* happen; but there are cases where, in spite of a smokeless stack, only a relatively small fraction of the heat contained in the coal is utilized while, on the other hand, even with considerable smoke making an extreme utilization of the coal can occur.

To explain these conditions more fully, I consider it proper, first of all, to recall the losses which take place during combustion.

When coal is burnt in a furnace a small quantity will always fall through the grate, likewise a small quantity will be inclosed by the coke. This coal is therefore withdrawn from combustion. Only in very rare cases are the flame-producing gases given off by the coal entirely consumed; on the contrary, these gases escape, more or less, without being consumed; generally a partial decomposition takes place during which carbon is separated in the form of soot. Consequently through these gases and vapors, partly visible and partly invisible, as well as through the soot, a certain quantity of heat is lost.

Of the actual heat developed, only a part is transferred to the heating surfaces as *useful*. A considerable portion escapes with the hot gases as free heat, therefore invisible, through the smoke pipe, while a further portion is lost by useless

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radiation from the boiler and brickwork or by conduction to surroundings.

For the conditions here outlined only the two losses through the stack come into consideration, namely, those due to the free heat in the waste gases and those due to imperfect combustion.

The first-mentioned loss is found by ascertaining the quantity of the waste gases, their temperature, specific heat and composition; also the composition of the fuel, in order that the amount of gas generated per kilogram may be calculated.

While, in general, the ascertaining of this loss can be accomplished without any special difficulty, unfortunately there has been lacking a method sufficiently accurate for the direct determination of the loss through imperfect combustion, for which reason this determination has always been avoided. This also accounts for the still very incomplete knowledge as to the relative amount of this heat.

As the heat lost through radiation likewise cannot be directly determined, it is combined with the loss due to imperfect combustion and designated as the residue in the heat balance.

Of all losses in a furnace, the one most apparent and disagreeably felt is that due to imperfect combustion. This, no doubt, is also the reason why those who are not better informed as to the conditions have given this loss the greater prominence.

Conditions being equal, there is no question but that, from an economical standpoint, imperfect combustion is more detrimental than a perfect one. The question is, how far is it possible to bring about perfect combustion and at the same time restrict the loss of free heat in the waste gases, and what importance is to be attached to the heat lost, economically considered, during practical working by imperfect combustion?

To answer this it will be necessary to get an idea of the loss of the waste heat and the circumstances influencing the same.

The loss of the waste heat naturally increases with the quantity and temperature of the waste gases. Therefore it becomes greater the greater the excess of air during combustion, since

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thereby the quantity of the hot gases naturally increases as well as their temperature, due to the conditions of heat conduction. The latter, moreover, becomes greater the more the boiler is forced.

Considering that 1 kg. of coal requires 7 to 9 cbm. of air, provided the process of combustion could be carried out with a quantity corresponding to its chemical composition, it is at once manifest that we have to deal with large gas volumes which in practice become still greater, as the above quantity is insufficient; it requires rather a multiple of this, depending upon the design of the furnace and the method of working adopted by the fireman. Moreover, for practical reasons, the temperature of waste gases below 180 to 200 degrees C. can no longer be utilized in a boiler.

If, for instance, combustion is effected with twice the volume of air, theoretically required (and even this does not indicate an unfavorable method of working) we have 16 cbm. of gas for 1 kg. of coal of average composition, supposing this reduced to a temperature of 0 degree C. and barometric pressure of 760 mm.

With a temperature of 250 degrees C. for the waste heat—which corresponds to moderate forcing for this method of working—there is in the heat lost about 1,180 T.U. for 1 kg. of coal, or, taking the heat value of the coal at 7,000 T.U., about 17 per cent. of the available heat.

This loss, entirely invisible to the eye, increases approximately in a direct ratio to a multiple of the minimum theoretical quantity of air and in direct ratio to the waste temperature. Cases where it reaches 30 per cent. are not in the least rare.

At any rate, from this it follows that the loss in the utilization of fuels is of considerable importance.

In opposition to this, what are the facts with regard to the loss through imperfect combustion?

As has already been said, the direct determination of this is very difficult. The only series of tests for its determination, based on a method of exact chemical measurement, date from

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the years 1879 to 1881, and were carried out by Bunte in the heat-test station at Munich.*

The main results of these tests were, that from waste gases containing about 8 to 10 per cent. of carbonic acid, therefore containing about 100 per cent. excess of air, very nearly perfect combustion could, in most cases, be obtained. A loss due to imperfect combustion of over 5 per cent. of the heat value of the coal occurred, in general, only in tests where the conditions of combustion were intentionally made unfavorable.

Furthermore, these tests show that imperfect combustion is more easily avoided the greater the excess of air, within the practical limits, but with it the waste heat also becomes greater, and that, generally, a considerable loss through imperfect combustion is only to be feared when the air supply is very limited.

Therefore, even though the latter loss can become quite considerable, the same importance is not to be attached to it for the utilization of fuels as that to the loss in waste heat, since according to the foregoing the loss through imperfect combustion moves, as a rule, contrary to that of the waste heat. As the loss through imperfect combustion, to any appreciable extent, generally results from an insufficency of air, a comparatively small loss of waste heat is connected therewith. If the obtaining of perfect combustion is conditional upon a larger supply of air, then the loss of the waste heat will become greater and the resulting gain is once more, at least partly, lost.

Especially on account of this circumstance technical circles, until recently, were inclined to attach less importance to the loss through imperfect combustion than it deserved.†

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*See H. Bunte, "Zeitsch, d. v. d. I.," 1900, p. 674; also "Bayerische Industrie and Gewerbeblatt," 1879 to 1881, H. Bunte: Report of the heat-test station, Munich.

† Relative to this, see also the discussion upon the test series given in Table V, where, despite an increasing loss through imperfect combustion. the utilization is improved in consequence of the simultaneous drop in the loss of the waste heat. In addition to this—as a further reason for unduly underrating the loss through imperfect combustion—is the fact that, according to the usual method of calculating the waste-heat loss, in the case of very imperfect combustion, it becomes too great, and with that the increase of the residue loss becomes too small compared with the residue loss obtained for perfect combustion. See the discussion on p. 645 at the conclusion of note.

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As to the conditions actually obtained, a series of trials which I carried out on a two-furnace boiler of standard make and which are to be discussed in the following, may furnish some information. The heating surface of this boiler was 75 square meters.

A clue to the amount of loss through imperfect combustion can be gained from the following method without the aid of an exact chemical analysis of the waste gases: Perfect combustion can be produced in a boiler by giving a proper supply of air and proper strength of draft; and by tests thus carried out, find the loss through radiation and conduction of the plant, from the residue in the heat balance, for the conditions of pressure in question.

Since this loss, with approximately equal pressure conditions on the boiler but different methods of working, will hardly vary, conclusions may be drawn—based upon a determination of the increase of the residue in the heat balance—through later tests to be made on the same plant, as to the loss due to imperfect combustion.*

The results of the tests carried out in this manner are given in Table I.

The construction and location of the boiler, which was in continuous service during the tests, can be seen in Figs. 1 to 3.

The four tests are not arranged in the order in which they were carried out, but according to the quantity of smoke developed and the quantity of air supplied. They were made between May 12 and 20, 1903, and the three designated I, II and III were each made twice under exactly the same conditions. The results of these duplicate tests agree with sufficient accuracy with one another. The duration of two tests was for a period of 9 and 9½ hours, respectively, and for all others, 10 hours.

The tests were made by taking accurate observations of all the conditions coming into consideration. The boilers for all the tests were attended by a fireman instructor.

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* Relative to this, see also p. 645, conclusion of note.

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		II.		
Heating surface of boiler, s				
Grate surface of boiler, squ Ratio of grate surface to he	Average.	May 14, 1903.	May 15, 1903.	Average.
Duration of trial, hours	19½	10	10	20
Fuel: English coal, West H	3,778	2,171	2,348	4,519
	193.5	217	235	226
	84 9	95.3	103	99.2
	2.47	2.77	3	2.88
Ashes, total, kilograms.....	337.5	264	247	511
in coal burnt, per ce	8.95	12.2	10.5	11.3
combustible consum	24 9	26.2	30.9	28.6
Feed water: total evaporati	26,412	13,620	14,800	28,420
evaporation pe	1,355	1,362	1,480	1,421
	17.26	17.35	18.85	18.1
average tempe	12.75	13	13	13
Steam: steam pressure, atm	6.76	6.76	6.8	6.78
average heat of eva	645	644.8	644.9	644.85
Gas analysis: (a) at end of	**13.4**	9.21	9.05	**9.13**
Carbonic acid, CO_2, averag	13.5	9.35	9.2	9.28
	5.17	9.6	10.35	9.97
Oxygen, O average of 15-m	18.67	18.95	19.55	19.25
Total of CO_2 and O, per ce	36	91	102	96
Excess of air (b) at end of b	9 32	6.4	6.7	6.55
	9.1	6 4	6.6	6.5
Carbonic oxygen, O, avera	9.8	12.8	12.8	12.8
Excess of air, per cent.....	93.5	183	171	177
Temperature, degrees cent	281	320	326	323
of air for com	22	22	23	22.5
Draft pressure, at the boile	5.65	12.6	12.6	12.6
Evaporation: (a) Water ev	6.99	6.27	6.3	6.29
(b) Equivalen at 0 degree	**7.08**	6.35	6.38	**6.37**

	U.	Per cent.	T. U.	Per cent.	T. U.	Per cent.	T. U.	Per cent.
Utilized for making steam..	13	69.25	4,043	59.9	4,063	62.1	4,053	61
Lost: (a) due to incomplet	81	2.75	258	3 8	264	4	261	3.9
(b) due to heat carrie	64	17.85	1,948	28.85	1,852	28.3	1,900	28.6
(c) other losses due t	57	**10.15**	502	7.45	365	5.6	434	**6.5**
Total = heat value of	15	...	6,751	...	6,544	...	6,648	...

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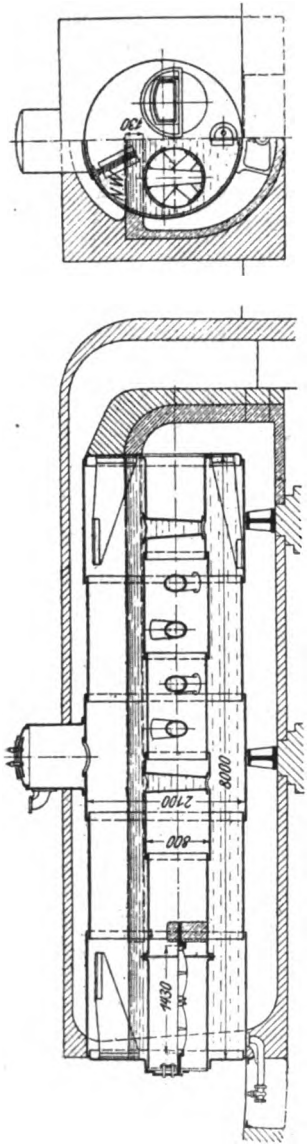

Figs. 1 and 2.

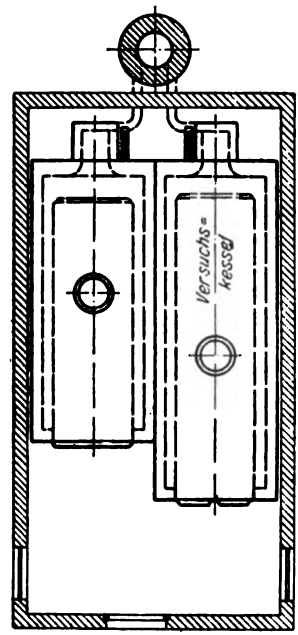

Fig. 3.

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The corresponding smoke diagrams are shown in Fig. 4.

In tests IV and I combustion ensued with an average proportion of 14.6 and 14.0 per cent. of carbonic acid at the end of the furnace. As they were very incomplete, therefore no conclusion can be drawn from them concerning the excess of air without a knowledge of the percentage of the unconsumed gases; * but at any rate it was very small. In test III the average proportion of carbonic acid formed at the end of the furnace was 13.4 per cent., and the excess of air about 35 per cent.; while for test II this excess was 100 per cent. for an average proportion of 9.1 per cent. of carbonic acid at the end of the furnace.

In the latter test the entire available stack draft was utilized amounting to about 13 mm. water column at the boiler end. The tendency was to produce as little smoke as possible, consequently to bring about perfect combustion without regard to heat utilization. The coal was thrown mainly along the center and more toward the front portion of the grate, so that plenty of air could be admitted along the sides and in the rear for combustion of the gases.

In test III good utilization of the fuel was to be obtained with as little smoke as possible. The fireman worked with a considerably reduced air supply, therefore a weaker draft, amounting to only 6 mm. of water column against 13 mm. as before. The coal was thrown upon the front part of the grate and after burning was carefully pushed back, especial care being taken to keep the grate well covered.

Test I was made with still less draft in the stack. Without any regard to the smoke, the best possible proportion of carbonic acid and the least possible excess of air was to be obtained. The coal was thrown upon the whole grate *uniformly* and the burning layer thereby kept thicker.

In test IV the excess of air was still more reduced.

The coal used was of English origin, of the West Hartley kind, containing about 27 to 29 per cent. of volatile matter,

* In relation to this, see also p. 645, conclusion of note.

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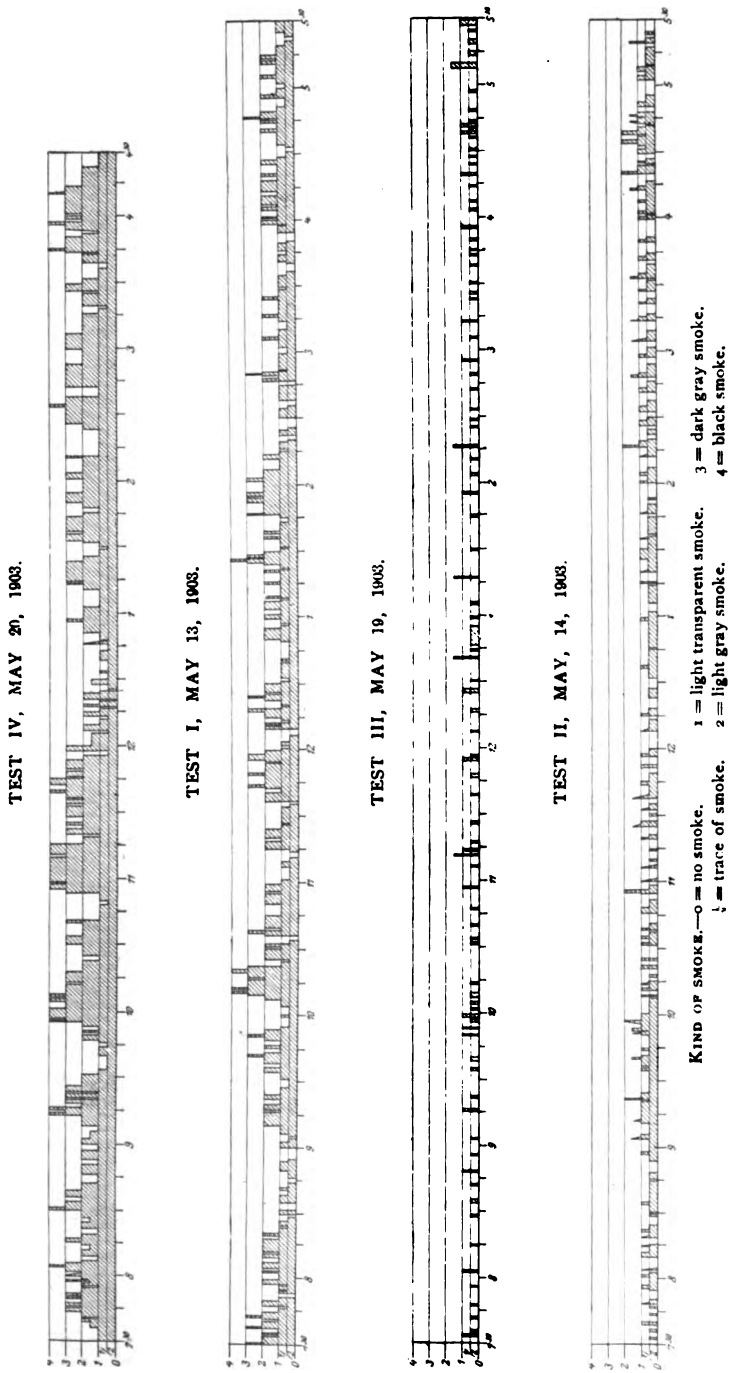

Fig. 4.

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therefore strongly inclined to smoke making. The average heat value was about 6,500 T. U.*

The hourly evaporation per square m. of heating surface was nearly the same on all days, averaging 16 to 18 kg., and can therefore be considered moderate. The coal consumption varied, according to the utilization desired, between 83 and 110 kg. per square m. of grate surface an hour.

VARIATION IN HEAT DISTRIBUTION, WITH THE AIR SUPPLY IN THE ORDINARY GRATE.

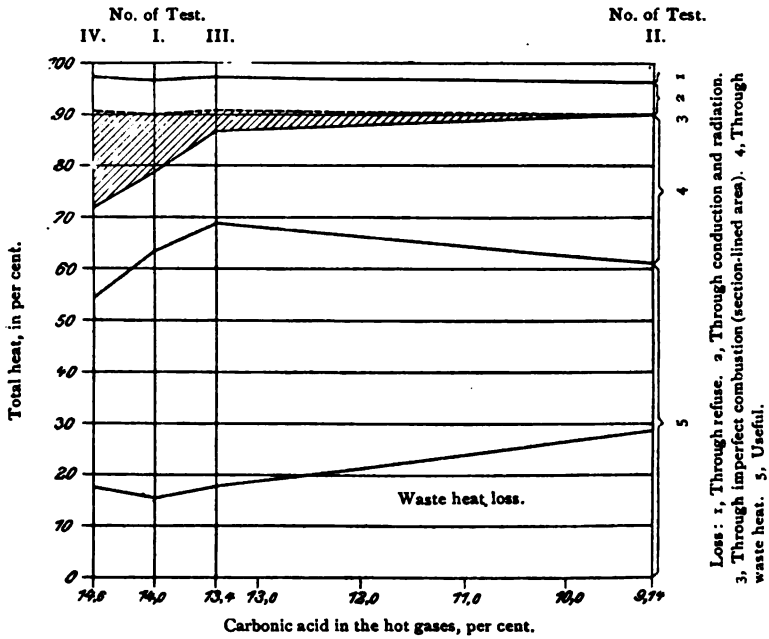

Fig. 5.

The graphical representation in Fig. 5 gives a very good idea of the results of the test. We see that the loss through imperfect combustion, which is represented by the shaded surface, can, indeed, obtain a not insignificant importance; but this first arises from conditions seldom occurring in practice, at least, not permanently. For operating under such condi-

*See Table III.

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tions (especially as they existed for test IV, and also for test I, weak draft at the boiler end, at the same time maintaining a heavy and uniform thickness of coal) is rather exceptional in practice. Further, it can be seen that the utilization was best when moderately smoking, and by increasing the supply of air the smoke could be thoroughly checked, but then the *utilization of the fuel*, in consequence of the great increase in the loss of free heat in the waste gases, decreased *rapidly* and became still less than in the case of test I, where imperfect combustion had already set in to a considerable degree. In the latter it still amounted to 63.5 per cent., compared with 61 per cent. during test II, with practically no smoke. But for *very* imperfect combustion, during test IV,* it fell rapidly below this figure to 54.4, while during test III it rose to 69.25† per cent. with moderate smoke making.

NOTE REFERRED TO IN TABLE I.

Regarding the heat balance, there is this to be said: The heat values have been recalculated on the basis of the quantity of pure ash established during the various tests. With the original values the distribution of the heat would have been as found in Table II. With respect to this, see the foot note to Table V.

The analyses of the coal samples are given in Table III.

The loss of free heat in the waste gases was calculated according to the formula as laid down in the standard rules for testing boilers and engines, viz:

$$V = \left(0.32 \frac{C}{0.536k} + 0.48 \frac{9H + W}{100}\right)(T - t) \text{ T. U.}, \quad . \quad (1)$$

* That imperfect combustion existed in a high degree during this test is evident from the after ignition observed in the upper flue, which continued for about five minutes, and during which the temperature at the end of the boiler rose to about 500 degrees C, the average observed being about 273 degrees C. Interesting also is the comparison of the total percentages of the carbonic acid and oxygen during the four tests, which, according to table I, averaged 17.05, 17.87, 18.67 and 19.25, which figures likewise indicate imperfect combustion with progressively decreasing air suppy.

† With respect to these figures it must be stated that the conditions of utilization during all of the tests have somewhat suffered, due to the porous condition of the brickwork around the boiler. This could not be altered for the tests, but it really does not prejudice the insight into the conditions under discussion.

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Table II.

Heat balance.	IV		I						III						II					
	T.U.	Per cent.	T.U.	Per cent.	T.U.	Per cent.	T.U.	Per cent.	T.U.	Per cent.	T.U.	Per cent	T.U.	Per cent.	T.U.	Per cent.	T.U.	Per cent.	T,U.	Per cent.
Utilized for making steam.....	3,497	56.5	4,165	63.6	4,206	62.9	4,186	63.25	4,514	69.4	4,511	70.6	4,513	70	4,043	61.2	4,063	65.1	4,053	63.15
Lost: (a) due to incomplete combustion of the carbon...	154	2 5	244	3.7	215	3.2	229	3.45	220	3 4	142	2.2	181	2.8	258	3.9	264	4.2	261	4.05
(b) Due to heat carried away in the chimney gases.....	1,090	17.6	978	14.9	1,062	15.9	1,020	15.4	1,102	17.	1,201	18.8	1,151	17.9	1,906	28.9	1,767	28.3	1,836	28.6
(c) Other losses due to conduction, radiation, soot, unconsumed gases, etc.....	1,450	**23.4**	1,163	17.8	1,199	18	1,181	**7.19**	664	10.2	533	8.4	599	**9.3**	398	6.	149	2.4	274	**4.2**
Total=heat value of fuel..	6,191	...	6,550	...	6,682	...	6,616	...	6,500	...	6,387	...	6,444	...	6,605	...	6,243	...	6,424	...

Table III.

Number of trial.....	IV	I		III		II		Average.	Variation.
Date of trial	May 20, 1903.	May 12, 1903.	May 13, 1903.	May 16, 1903	May 19, 1903.	May 14, 1903	May 15, 1903.		
Carbon, per cent	65.57	69.12	69.70	68.42	67.50	69.27	66.03	.....	
Hydrogen, per cent	4.30	4.74	4.76	4 61	4 68	4.58	4.36	.....	
Oxygen and nitrogen, per cent.....	8.72	9.48	9.54	9.78	9.66	8.55	9 26	.....	
Sulphur, per cent.....	1.71	.84	.91	1.17	1.13	1.01	1.14	.....	
Ash, per cent	9.17	8.78	8 36	7.24	7.28	10.02	10.51	8.77	
Moisture, per cent.....	10.53	7.04	6.73	8.78	9 75	6.57	8.70	.....	
Volatile matter, per cent.....	27.12	28.9	27.6	28.45	26 81	27.55	26.9	.....	
Fixed carbon, per cent	53.18	55 28	57 31	55.53	56.16	55.86	53.89	.....	
Heat value of coal, T.U..	6,191	6,550	6,682	6,500	6,387	6,605	6,243	6,451	+3.58 p.c. −4.03 p c.
combustible, T.U.....	7,788	7,831	7,917	7,803	7,768	7,966	7,792	7,838	+1.63 p.c. − .89 p.c.
Percentage of ash in total coal consumed, per cent.....	6.06	8.36	9.29	6.13	6.57	8.19	6 65	7.33	.....
Heat value of combustibles, T.U.....	6,433	6,583	6,608	6,587	6,442	6,751	6,544	6,564	+2.85 p c. −2. p.c.

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in which the part contained in the refuse and withheld from combustion has been deducted from the total carbon.

This formula was criticised by Eberle in the "Journal of the Bavarian Steam Boiler Inspection Society," of 1902, pp. 25 and 26, where he proves that it is only strictly valid in case the velocity of combustion varied proportionately to the amount of carbonic acid, while for a uniform velocity of combustion the calculation should be made according to the following formula

$$V = \frac{a}{n} \Sigma \left(\frac{T - t}{k} \right) + \frac{b}{n} \Sigma (T - t)$$

in which

$$a = 0.32 \frac{O}{0.536}, \; b = 0.48 \frac{9H + W}{100} \quad . \quad . \quad . \quad . \quad (2)$$

and n the number of simultaneous observations of the percentage of carbonic acid and temperature of the waste gases.

Eberle concludes from his deductions that the calculation of the loss of the waste heat from the temperature and the percentage of carbonic acid of the waste gases can only be regarded as reliable if the velocity of combustion is uniform, and then the calculation should be according to formula (2).

For the present tests the values found from both formulae are given in Table IV.

Table IV.

Number of trial	IV	I		III		II	
Date of trial	May 20, 1903.	May 12, 1903.	May 13, 1903.	May 16, 1903.	May 19, 1903.	May 14, 1903.	May 15, 1903.
Loss of heat in waste gases, T.U.:							
According to formula (1)	1,128	983	1,051	1,116	1,211	1,948	1,852
According to formula (2)	1,172	1,015	1,100	1,153	1,233	2,078	1,936
Difference:							
T. U	44	28	49	37	22	130	84
In per cent. of the heat value of the coal	0.7	0.42	0.74	0.56	0.34	1.93	1.27

As is evident, the variations are slight, and for the large loss of 30 per cent. in waste heat during test II attain values of

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over 1 per cent., viz: 1.27 and 1.93 per cent., respectively, of the heat value of the coal.

Now, before heat tests can have any claim to accuracy, without doubt one's efforts, for other good reasons, would be to establish equal conditions. In the present case it was endeavored to attain this if possible where, of course, slight fluctuations in the duty, due to the respective conditions of management, could not be entirely avoided. Even if complete uniform duty could be established and all other conditions coming into consideration remain invariable, so far as this is possible, still approximately equal velocity of combustion could only be obtained in furnaces by continuous stoking (*e. g.* by mechanical stoking, inclined grates and similar constructions). But then, at the same time, the influence of the fluctuations in carbonic acid and of the temperature of the waste gases would be so slight that any variation obtained from the two formulae for the waste heat loss will hardly exceed the limits of accuracy of the determination.

In the ordinary grate, on the contrary, when periodically stoked and when using fuels very rich in gas, it is generally impossible, though all other conditions remain the same, to obtain a velocity of combustion which does not vary. This will rather vary from one period of stoking to another in such a manner that it will increase rapidly, immediately after stoking, with the ignited and freshly-thrown on coal, and will then again gradually decrease with the diminishing gas abstraction. That this is a fact is already evident from the amount of air required for such furnaces, in case perfect combustion is to be obtained, being generally greater after than before stoking.

The fluctuation, being conditioned by the more intense combustion of the volatile matter in the fuel, will become greater the more gas the coal contains and the more rapid the abstraction of gas.

If no special regulation of the air supply takes place, the percentage of carbonic acid in the heat gases will be found considerably greater during the abstraction of gas than after the completion of the same, so that, as a matter of fact, it

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varies in a similar manner as the velocity of combustion, being calculated according to formula (1), therefore in this case at least just as much qualified as formula (2).

On the other hand, if less fluctuation in the percentage of carbonic acid is obtained by a proper regulation of the air supply, then the variation in the calculation according to both formulae will be insignificant.

Hence if proper care is taken to maintain nearly uniform conditions during a test, the determination of the waste heat loss according to formula (1), therefore based upon average values, will be completely within the limits of accuracy even for the ordinary grate,—naturally presupposing that carbonic acid tests are frequently made.

Concerning the waste heat losses and the residue not accounted for in the heat balance, it may be said, in the case of very imperfect combustion (as for example prevailed during tests IV and I), that the waste heat loss obtained, according to the usual method of calculating, is too large; the residue loss, therefore, becomes too small, so that from a comparison of the residue losses thus obtained the loss through imperfect combustion, as represented in Fig. 5, will be too small rather than too large. For such conditions the excess of air and volume of the gases of combustion obtained will likewise be too large if calculated according to the usual method for perfect combustion.

If, in the case of imperfect combustion, the combustible in the waste gases be determined according to the method practiced by Bunte for heat tests in the heat-test station at Muenich (a method quite recently improved by him), whereby it can be assumed, without any appreciable error, that all carbon as carbon oxide, and all hydrogen, exist in an independent state, a more accurate value for the waste heat loss can be calculated as follows:

Let C represent the fraction of the carbon contained in 1 kg. of coal to be burnt. Further, let c_1 be the loss of carbon per kg. in the refuse, c_2 the corresponding loss in unconsumed gases, and c_3 that of the soot, and let k_1 represent the percentage

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of carbonic acid, k_2 the percentage of carbon oxide, h the percentage of hydrogen, and r the grammes of soot found in 1 cbm. of dry gas. Since during the combustion of 1 kg. of carbon in pure oxygen $\frac{1}{0.536}$ cbm. of carbonic acid is generated (referred to 0 degree C. and 760 mm. barometric pressure), then, if V_g be the volume of dry gas generated from 1 kg. of coal, there results the following relation for the same:

$$\frac{C - c_1 - c_2 - c_3}{0.536} : V_g = k_1 : 100$$

or

$$V_g = 100\left(\frac{C - c_1 - c_2 - c_3}{0.536 k_1}\right) \quad . \quad . \quad . \quad . \quad (1)$$

In this expression c_1 is known. If R kg. of refuse with v per cent. of combustible were found after the combustion of B kg. of coal, we have

$$c_1 = \frac{v}{100}\frac{R}{B} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

For c_2 and c_3, we have the relations

$$\frac{c_2}{0.536} = \frac{k_2}{100} V_g. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

and

$$c_3 = \frac{r}{1{,}000} V_g \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The two latter expressions substituted in formula (1) give

$$C - c_1 - 0.536\frac{k_2}{100} V_g - \frac{r}{1{,}000} V_g = 0.536\frac{k_1}{100} V_g$$

or

$$V_g = \frac{C - c_1}{0.536\left(\frac{k_1}{100} + \frac{k_2}{100}\right) + \frac{r}{1{,}000}} \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Further, let H be the fraction of hydrogen, and W the fraction of moisture per kg. of coal, and if H_1 is the loss of

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hydrogen in kg. referred to 1 kg. of coal, then the steam generated from 1 kg. of fuel will be

$$G_w = [9(H - H_1) + W]\text{kg.} \quad . \quad . \quad . \quad . \quad (6)$$

in which

$$H_1 = 0.0896 \frac{h}{100} V_g \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

Then from the specific heat c_g per cbm. of gas and c_w per kg. of steam the waste heat loss found will be

$$V_1 = \left\{c_g \frac{C - c_1}{0.536\left(\frac{k_1 + k_2}{100}\right) + \frac{r}{1{,}000}} + c_w [9(H - H_1) + W]\right\} (T - t) \text{ T. U.}, \quad . \quad . \quad (8)$$

in which

$$H_1 = 0.0896 \frac{h}{100} V_g.$$

On the other hand, the loss through imperfect combustion found will be

$$V_2 = \left[\frac{k_2 + h}{100} V_g \cdot 3{,}000 + r V_g \times 8.1\right] \text{T. U.} \quad . \quad (9)$$

as the heat value of 1 cbm. of carbonic oxide as well as of 1 cbm. of hydrogen can be taken at about 3,000 T. U. and that of 1 gramme of soot at 8.1 T. U.

In Tables V and VI are compiled the results of a further series of tests, which are also very instructive.

These tests were made on a battery of five 2-furnace flue boilers, containing a total of 470 square meters of grate surface. The boilers were in continuous operation, and exhibited good performance conditions. The make and the arrangement of the boilers are shown in Figs. 6 and 7.

From a thorough examination of the waste gases, it was determined that the boilers were operated with too large an excess of air, in consequence of which the loss in free heat in the waste gases amounted to from 28 to 29 per cent. of the

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Table V.

Boiler	I	II, III	IV, V
Heating surface, in contact with water, sq. m.	103	84.7	99.1
total sq. m.	120	110.2	123.2
Grate surface, tests 1 to 4 sq. m.	2.98	2.8	2.72
test 5 sq. m.	2.69	2.69	2.54

	Trial 1, March 4, 1903.	Trial 2, March 5, 1903.	Average.	Trial 3, March 6. 1903.	Trial 4, March 7, 1903.	Average.	Trial 5, June 18, 1903.
Duration of trial, hours	10	10	20	10	10	20	10
Total heating surface, in contact with water, square meters	470.6	470.6	470.6	470.6	470.6	470.6	470.6
grate surface, square meters	14.02	14.02	14.02	14.02	14.02	14.02	13.15
Ratio grate surface to heating surface	1 : 33.6	1 : 33.6	1 : 33.6	1 : 33.6	1 : 33.6	1 : 33.6	1 : 35.8
Fuel: English coal, Real West Hartley main:							
Total coal burned, kilograms	10,615	10,562	21,177	9,532	10,728	20,260	11,135
per hour, kilograms	1,062	1,056	1,059	953	1,073	1,013	1,114
square meter of grate surface	76	75	75.5	68	77	72	84
heating surface	2.26	2.24	2.25	2.03	2.28	2.15	2.36
Ashes, total, kilograms	938	943	1,881	775	839	1,614	919
in coal burned, per cent	8.8	8.9	8.85	8.1	7.8	8	8.2
combustible (carbon), per cent	21.1	14.8	18	25.5	22.7	24.1	16.3
Feed water: total evaporation, kilograms	79,791	80,705	160,496	79,903	88,952	168,855	88,178
evaporation per hour, kilograms	7,979	8,071	8,025	7,990	8,895	8,443	8,818
sq. m. of heating surface, per hour, kilo.	17	17.2	17.1	17	18.9	17.9	18.7
average temperature, degrees centigrade	57.1	67.6	62.3	65.3	69.1	67.2	58.1
Steam: pressure, boiler I, atmospheres	6.7	6.6	6.65	6.5	6.7	6.6	6.5
boilers II to V, atmospheres	11.6	11.6	11.6	11.65	11.55	11.6	11.7
average heat of evaporation, T. U.	605.8	595.2	600.5	597.4	593.8	595.6	604.7
Heat gases: carbonic acid at the end of boiler, per cent	**8.7**	**8.5**	**8.6**	**12.6**	**12.7**	**12.6**	**10.5**
excess of air at end of boiler, per cent	112	117	115	46	45	46	76
temperature at end of boiler, degrees centigrade	**350**	**353**	**352**	**309**	**318**	**314**	**356**
of the air, degrees centigrade	18	20	19	23	23	23	27
Draft pressure at end of boiler, millimeters of water column	8.3	7.2	7.8	2.3	2.1	2.2	4.9
Evaporation: (a) Water evaporated per kilogram of fuel, kilograms	7.52	7.64	7.58	8.38	8.29	8.33	7.92
(b) Equivalent evaporation from and at 100° C. per kilogram of coal from water at 0° C., kilograms	7.15	7.14	7.14	7.86	7.73	7.79	7.52

Heat balance.*	T. U.	Per cent.	T. U.	Per cent.	T. U.	Per cent.	T. U.	Per cent.	T. U.	Per cent.	T. U.	Per cent.	T. U.	Per cent.
Utilized for making steam	4,556	67.1	4,547	67	4,552	**67.05**	5,006	72.1	4,923	71.4	4,964	**71.75**	4,789	**72.3**
Lost: (a) in the refuse	152	2.2	107	1.6	129	1.9	168	2.4	144	2.1	156	2.25	109	1.6
(b) due to heat carried away in the chimney gases	1,690	24.9	1,715	25.3	1,702	**25.1**	1,048	15.1	1,055	15.3	1,051	**15.2**	1,384	**20.9**
(c) remainder=conduction, radiation, soot, unconsumed gases, etc.	389	5.8	412	6.1	400	**5.95**	722	10.4	777	11.2	749	**10.8**	343	**5.2**
Total = heat value of fuel	6,787	...	6,781	...	6,784	...	6,944	...	6,899	...	6,921	...	6,625	...

* See note page 658.

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Table VI.—DATA TAKEN AT THE DIFFERENT BOILERS.

Number of trial.	Number of boiler.	Coal consumed. Total.	Coal consumed. Per hour.	Coal consumed. Per hour per one square meter of grate surface.	Coal consumed. Per hour per one square meter of heating surface.	Per cent. of carbonic acid at the end of boiler.	Excess of air at the end of boiler.	Temperature. Chimney gases.	Temperature. Air for combustion.	Draft at the boiler end.	Loss in free heat carried away in the chimney.
		kilos.	*kilos.*	*kilos.*	*kilos.*	*pr. ct*	*pr. ct.*	° *C.*	° *C.*	*mm. W. C*	*pr. ct.*
Trial 1.	I	2,203	220	74	2.13	8.4	120	327	...	10.2	24.1
	II	2,300	230	82	2.72	9.4	96	364	...	7.6	24.2
	III	2,209	221	79	2.61	10	85	376	18	7.5	23.4
	IV	1,970	197	72	1.99	8.3	122	362	...	7.6	27
	V	1,933	193	71	1.95	7.4	150	321	...	8.4	26.2
Trial 2.	I	2,144	214	72	2.08	8.6	114	326	...	8.6	23.3
	II	2,181	218	78	2.58	9	105	364	...	6.1	23.9
	III	2,158	216	77	2.55	9.8	88	385	20	6.2	24.9
	IV	2,094	209	77	2.11	8	130	367	...	7.4	28.4
	V	1,985	199	73	2	7.2	156	322	...	7.8	26.5
Trial 3.	I	2,166	217	73	2.1	11.5	60	308	...	3.6	16.4
	II	1,776	178	64	2.1	12.9	43	307	...	3.4	14.7
	III	1,681	168	60	1.99	14.1	30	324	23	1.1	14.5
	IV	1,952	195	72	1.97	13.1	40	313	...	1.5	14.7
	V	1,957	196	72	1.97	11.6	59	292	...	1.8	15.3
Trial 4.	I	2,213	221	74	2.15	12.1	52	302	...	3	15.1
	II	2,440	244	87	2.88	13	42	332	...	2.9	15.8
	III	1,930	193	69	2.28	13.7	34	335	23	1.3	15.3
	IV	2,174	217	80	2.19	13.6	35	327	...	1.8	14.9
	V	1,971	197	72	1.99	11.1	66	295	...	1.5	15.7
Trial 5.	I	2,479	248	92	2.41	10.4	78	359	...	6.7	21.2
	II	2,057	206	77	2.43	11.2	65	370	...	2.8	20.5
	III	2,479	248	92	2.93	10.6	74	404	27	4.7	23.8
	IV	2,072	207	81	2.09	10	85	344	...	4.7	20.9
	V	2,048	205	80	2.07	10.1	83	301	...	5.7	17.9

heat value of the coal, for which reason these tests were undertaken. It was decided to ascertain how far the excess of air could be restricted, and thus determine how much the utilization could be raised. The large excess of air was partly due to the fact that the grates were too long, which made it difficult to keep the back covered. Before starting the tests the grates were shortened. The furnaces were equipped with the Kowitzke upper-air supply, the air being delivered through the bridge wall. The duration of each test was ten hours. Dur-

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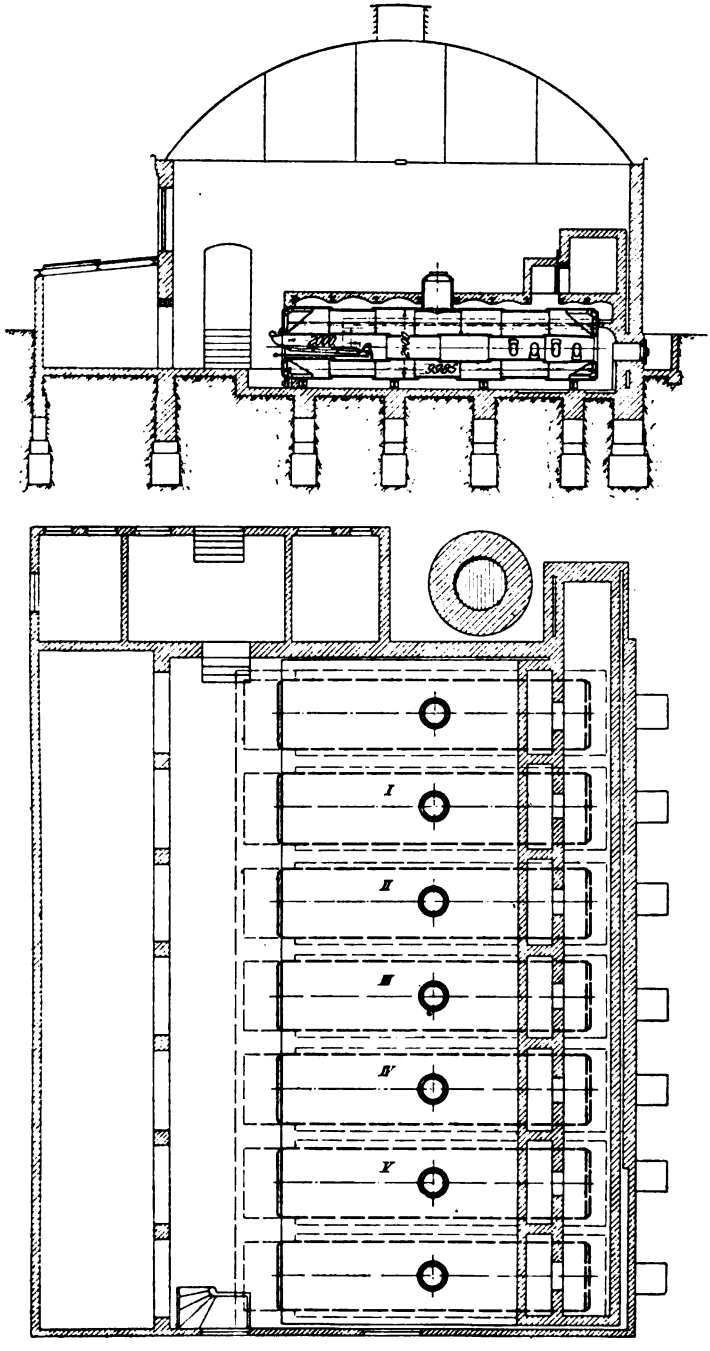

Figs. 6 and 7.

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ing the first two days of the tests the boilers were attended by the regular firing force from the factory. The excess of air compared with the original examination had somewhat decreased (as the boilers were operated under a pretty strong draft; moreover, the firemen stoked badly and did not permit the fires to burn down uniformly), but the loss in free heat through the smoke stack, according to the heat balance, still amounted to about 25 per cent. The utilization of the fuel was about 67 per cent., and in the refuse was found 1.9 per cent., leaving 6 per cent. as a residue. Since the development of smoke was moderate, it may be assumed that the residue loss of 6 per cent. is mainly to be credited to conduction and radiation. The residue loss can be considered as moderate, which is to be attributed to the good condition of the brickwork and lagging and to the small amount of radiating surface exposed by the five adjacent boilers.

I wish to add that, for the purpose of determining the waste heat losses as accurately as possible, the percentages of carbonic acid and the temperatures of the gases were taken every three or four minutes, at the end of each boiler. This was also done in the case of the first series of tests.

For the purpose of establishing the conditions that would obtain, if the method of working was changed, it was decided to let two firemen instructors take charge of the installation on the third and fourth days. The method of working was to be changed in such manner that the average per centage of carbonic acid in the waste gases obtained on the first and second days of the trials was to be increased from 8.6 to 12 or 13 per cent., or the excess of air of about 115 per cent. obtained on the first two days was to be cut down to about 40 or 50 per cent. To obtain perfect combustion of the gases under these conditions (the coal used contained about 30 per cent. of volatile matter) it was found that the grates were still too large despite the fact that they had already been shortened. To bring about the reduced excess of air the boilers had to be operated under a very light draft, and, although the doors for the supply of secondary air were constantly kept open, the

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formation of smoke increased considerably. The comparison of the heat balances is very instructive. As a result of cutting down the supply of air, and in connection with this the drop in the gas temperatures at the end of the boilers, the loss in free heat in the waste gases fell from 25.1 to 15.2 per cent. —very nearly 10 per cent.—but at the same time very imperfect combustion was obtained. In consequence of this the residue loss in the heat balance was increased from about 6 to 11 per cent., therefore about 5 per cent. of the heat value of the coal was lost due to a more intense smoke formation.* But against this loss there was a falling off in the loss of the waste heat of nearly 10 per cent., so that in spite of the formation of smoke, which was considerable, the factor of utilization increased from about 67 to about 72 per cent.

Of course, matters were not permitted to remain thus: after the conditions had been cleared up, the cast-iron bridge walls were shifted and the grates still further shortened.

About three months later a further series of tests were made under the same conditions. It was found that the boilers could now be operated with a stronger draft, so that, for an average of 10.5 per cent. of carbonic acid in the waste gases, there was no excessive smoke developed. In spite of the fact that the boilers were not in as clean a condition as during the former tests (the waste-gas temperatures were relatively higher), the utilization obtained amounted to 72.3 per cent. The loss in free heat in the waste gases was 20.9 per cent., and the loss in the refuse, as well as in the residue, were about the same as originally. These losses are given in the last column of Table V.

In Table IX are given the results obtained in *water-tube boilers* with lignite, a fuel that is extraordinarily rich in gas.

As is well known, these boilers often cause the formation of very dense smoke, particularly with fuels very rich in gas, and therefore, in case the furnace is not well designed, they can cause fairly large losses through imperfect combustion.

*In regard to this see the remarks at the conclusion of note on p. 641, according to which the loss in waste heat was really still less and the increase of the loss through imperfect combustion was correspondingly greater.

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Table IX.

Group.	I.		II.		III.		IV.		
					Ordinary grate.		Step grate with secondary air supply.		
							Steam generated in		
	Ordinary grate.	Step grate with secondary air supply.	Ordinary grate.	Step grate with secondary air supply.	Lignite.	½ Ruhr coal. ½ Lignite.	One boiler.	Two boilers.	
Heating surface, square meters	160	160	153	153	235	235	200	2×200=400	2×200=400
Grate surface, square meters	3.182	2.98	3.50	2.84	4.95	4.95	5.06	2×5.06=10.1	2×3.57=7.14
Steam generated per square meter of heating surface, kilograms	11.0	13.0	19.4	16.5	12.5	14.9	15.1	7.23	7.53
Coal consumption per square meter of grate surface, kilograms	128	188	181	143	167	99	121	50.8	73.6
Carbonic acid, per cent	9.5	7.2	11.3	9.7	9.7	7.8	14.5	8.6	7.7
Temperature of the waste gases, °C	371	341	405	284	376	311	374	267	250
Heat value, T. U	5,442	4,121	5,425	5,768	4,934	6,740	5,180	5,180	5,040
Heat utilized, per cent	50.5	57.5	55.1	68.9	45.9	67.5	62.2	70.2	72.1
Waste heat loss, per cent	25.3	31.6	23.3	18.2	26.1	25.7	17.4	19.3	19.7
Ashes, per cent	7.9	7.7	0.9	4.4	2.0	1.0	0.4	0.3	0.2
Residue loss, per cent	**16.3**	**3.2**	**20.7**	**8.5**	**26.0**	**5.8**	**20.0**	**10.2**	**8.0**

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The figures in the first three columns are taken from an article by Eberle and Reischle, entitled "The main results from an engineering standpoint of the activity of the Bavarian Association for the Inspection of Steam Boilers," published in the "Zeitsch. d. Vereines d. Ing.," 1903, page 1039. These figures are of such value and interest that they are reproduced here. In tests I and II, the point in question was to replace the ordinary grates under such boilers by step-grates having an upper-air supply, according to the Dürr system. The figures in the third column are derived from two tests, both of which were carried out with the ordinary grate. One test was made with lignite, and the other with a mixture of $\frac{2}{3}$ Ruhr coal and $\frac{1}{3}$ lignite—a fuel very much poorer in gas.

In the article mentioned, the following remarks were made concerning these tests:

"By comparing only the residue losses in this table—in which, besides the losses through radiation and conduction, are also contained the losses through soot and unconsumed gases, as well as possible errors in observation—it can be seen that, through the substitution of the step-grate with a secondary air supply for the ordinary grate in tests I and II, the combustion, as well as the heat utilization, was materially improved. The same thing was obtained in test III by using a mixture of $\frac{2}{3}$ Ruhr coal and $\frac{1}{3}$ lignite, instead of all lignite, as used in the first test. In this test the residue loss was reduced from 26 to 5.8 per cent."

Further it is stated:

"If water-tube boilers are to be used for burning fuels rich in gas, as, for example, lignite, then in selecting the furnace arrangement especial care must be taken that the combustion is complete in the fire space, for, after the hot gases have once entered the tubes, further combustion is almost stopped, due to the fact that the gas stream has considerably cooled off and split up."

Similar results are shown in column 4, Table IX. These results are derived from the heat balance of tests taken from the "Journal of the Bavarian Association for the Inspection of

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Boilers," November 15, 1903. The tests were made on two water-tube boilers of the same size, equipped with step-grates and an upper-air supply. One test was made with one boiler in use, and the other test with two boilers in use, the steam generation being the same in both cases. The fuel used was Bohemian lignite. The performance in the first test was 15.1 kg. of water evaporated per square meter of heating surface per hour, and in the second test, 7.23 kg. and 7.53 kg., respectively. The utilization in the test with two boilers amounted to 70.2 and 72.1 per cent., respectively. The report on the test says:

"The result of the test with one boiler with twice the steam performance was materially less satisfactory, since only 62.2 per cent. of the heat was utilized. But the falling off in the utilization in this test is to be attributed solely to imperfect combustion of the coal. Despite the fact that all openings for the supply of air were kept open and the draft in the stack was of the best, it was impossible to supply the quantity of air necessary for perfect combustion, the result being that the carbonic acid averaged 14.5 per cent.—which might be considered satisfactory, but at the same time there was a considerable loss in the residue, amounting to 20 per cent., one-half of which is probably due to the unconsumed particles in the hot gases. This result could be foreseen by the heavy emission of smoke from the stack during the tests."

From this it can be seen that, when burning fuels very rich in gas in water-tube boilers, there may be considerable losses if the furnace is not a suitable one for the boiler, or if the grate consumption is too high.

Returning once more to the first series of trials (Table I), test II shows distinctly that it is not a very difficult matter to operate boilers (at least in the case of flue boilers) having ordinary grates with little smoke emission, even for coal very rich in gas and a grate consumption up to 100 kg., as long as *no value* is placed upon the utilization of the fuel, and that perfect combustion—therefore smokeless combustion—can also exist without good utilization of the coal.

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It will be understood that, so far as the latter point does not enter into the question, care need only be taken to provide a supply of air sufficiently large in the combustion space, under a good draft, so that it reaches the gases before they have been cooled below the temperature necessary for their ignition.

Moreover, the initial temperature (at least between certain limits), with regard to the formation of smoke, is of considerable less influence than a lack of air after stoking, as is seen from test II, where, as a result of the large excess of air, the temperature in the fire space doubtless was at its lowest; nevertheless the formation of smoke was least, due to the fact that the gases quickly obtained the air necessary for combustion That it was not a question of smoke abatement in this case, but one of more perfect combustion, is shown by the loss in the residue.

The foregoing statement is further verified by the series of tests in Tables I and V, which show that the same importance is not to be attached to the loss through imperfect combustion, for the utilization of fuels, as that to the loss in free heat in the waste gases, even though imperfect combustion, in certain cases, can undoubtedly assume an importance not to be neglected.

As already mentioned, in the practical operation of boilers (especially *flue boilers*), such large losses through imperfect combustion, as were intentionally brought about in the first named series of tests, are exceptional. The reason for this is that, as a rule, smoke development usually occurs only for a short period after the stoking and working of the fires—and especially when the fireman permits the fires to get too low, then upon stoking throws in too much coal and imprudently stirs the fire. If in such a case, by working more carefully, not only a decrease in smoke formation occurs, but the utilization itself can also be considerably raised, then the latter is due to the fact that the waste heat loss becomes less.

With the method of firing—frequently met with—there arises, besides the temporary intense smoke development, generally a large excess of air, and with this a heavy loss in waste heat. The formation of smoke is of course brought

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about through a lack of air, which only continues during the very rapid abstraction of the gas from the fuel. After the termination of the gas abstraction, the quickly formed vacant places on the grates produce a considerable excess of air, and in this way cause an increase in loss through waste heat.

The increasing of the utilization by adopting a better method of working is therefore not to be attributed—as the layman is inclined to accept—to the difference in loss through imperfect combustion, but is mainly due to the variation of the losses in free heat in the waste gases.

Therefore these tests and discussions show that the development of smoke is not, as a rule, accompanied by a large loss in fuel, as is generally supposed by those who are not familiar with the question. *Undoubtedly as a result of a lack of understanding of the principles involved in the management of furnaces, vast quantities of heat are lost, so that it well pays to give more consideration to this than has generally been the case.* The losses are not so much to be sought in the smoke, as in the invisible escaping waste heat of our furnaces. If, therefore, by the abatement of smoke development economic interests are not directly affected to the extent frequently supposed—the question generally being one of public interest—nevertheless both questions, the abatement of smoke and extreme utilization of fuels, are so intimately connected that, to obtain a wholesome progress, it will not be well to treat one without the other.

Smokeless working is, in general, more difficult the more we attempt to obtain economical combustion, hence, the more we strive to restrict the excess of air. These very conditions, therefore, constitute the *pith of the whole smoke question*, and in them lies the difficulty of the solution of the smoke question, if this is to be obtained in a simple manner, guaranteeing certainty in management and at the same time good utilization of fuel.

It is not sufficient to furnish a supply of air in keeping with the demand, but this supply must be so regulated that the minimum amount of air may be thoroughly mixed with the

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gases before they have had time to cool down too much. We must not, for the sake of economy, demand the maximum decrease in smoke formation, but should rather be contented with a condition to be designated as *light smoke.*

NOTE REFERRED TO IN TABLE V.

These heat balances are based upon the heat values as found by the recalculation from the percentage of ash determined from average samples taken during the tests. The distribution of heat obtained from the original values is shown in Table VII.

The reason for making a recalculation was due to the large variations in the utilization and residue loss, as seen in Table VII—obtained from the duplicate tests I and II, as well as from those of III and IV—in spite of the slight differences in the factors of evaporation, and in the number of heat units utilized per kilogram, based upon the heat values of the samples.

There is undoubtedly ground for objecting to the recalculation, inasmuch as a complete and certain determination of the percentage of pure ash may be impracticable, a part of the ash being carried along in the flues. For fuels producing much, and specifically light ash, such as lignite, the amount of ash carried along in the flues will be considerable and hence would prohibit such a recalculation. However, in the present case, where the boilers were operated partly under a very light draft, the error in the determination of the percentage of ash could not have been large. Furthermore, the error would have to manifest itself similarly for all tests; consequently, *as a comparison for the residue losses*, which is the point in question here, the error would disappear. Very careful attention was, of course, given that the process of cleaning fires was effected at exactly the same time before beginning and before terminating each test. There was also no fluctuation of the load on the boilers during the various tests.

Therefore, the reason for the recalculation in the present case, where about 10,000 kilograms of coal were consumed dur-

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Table VII.

Heat balance.	Trial 1.		Trial 2.		Average.		Trial 3.		Trial 4.		Average.		Trial 5.	
	T. U.	Per ct.	T. U.	Per ct.	T. U.	Per ct.	T. U.	Per ct.	T. U.	Per ct.	T. U.	Per ct.	T. U.	Per ct.
Utilized for making steam	**4,556**	67.5	**4,547**	64.8	4,552	66.1	**5,006**	71.9	**4,923**	75.2	4,965	73.5	4,789	72.4
Lost: (a) in the refuse	152	2.2	107	1.5	129	1.85	168	2.4	144	2.2	156	2.3	109	1.6
(b) due to heat carried away in the chimney	1,681	24.9	1,773	25.2	1,727	25 05	1,051	15.1	1,005	15.4	1,028	15.3	1,384	20.9
(c) other losses (residue) due to conduction, radiation, soot, unconsumed gases, etc	364	5.4	594	8.5	479	7	737	10.6	471	7.2	604	8.9	337	5.1
Total heat value of the fuel	6,753	...	7,021	...	6,887	...	6,962	...	6,543	..	6,753	...	6,619	...

Table VIII.

Number of trial	I		II.	III		IV.	Average of I to IV.	Variation per cent.	V	
Date of trial	March 4, 1903.		March 5, 1903.	March 6, 1903.		March 7, 1903.			June 18, 1903.	
.....	Sample 1.	Sample 2.	...	Sample 1.	Sample 2.	...			Sample 1.	Sample 2.
Carbon, per cent	70.2	72.18	73.9	74.2	72.63	68.4	...	...	70.43	70.04
Hydrogen, per cent	5.6	4.66	5.4	5.1	4.82	4.1	...	...	4.7	4.84
Oxygen and nitrogen, per cent	...	8.54	...	...	8.38	...	...	...	8.79	9.48
Sulphur, per cent	...	1.68	...	...	1.68	...	...	...	1.35	0.95
Ash, per cent	**9**	**5.88**	**4.6**	**5.3**	**6.16**	**10.5**	**6.99**	...	6.23	6.28
Moisture, per cent	7.5	7.06	7.1	7.3	6.33	7.1	...	...	8.5	8.41
Volatile matter, per cent	30.8	30.32	31.5	30.5	30.62	28.6	...	...	30.42	30.44
Fixed carbon, per cent	52.7	56.74	56.8	56.9	56.89	53.8	...	...	54.85	54.87
Heat value of the coal, T. U.	6,643	6,862	7,021	6,987	6,936	6,534	6,830	+2.8 −4.3	6,693	6,658
Heat value of combustible matter, T. U.	8,000	7,930	**8,000**	8,044	7,969	**7,993**	**7,989**	+0.69 −0.74	7,909	7,864
	7,965			**8,006**					7,886	
Ashes in total coal consumed, per cent	**6.96**		**7.6**	**6.1**		**6.05**	**6.68**	...	6.9	
Heat value on the basis of ash in total coal consumed, T.U.	6,787		6,781	6,944		6,899	6,853	+1.33 −1.05	6,625	

Samples 1 and 2 from the trials of March 4 and 6, 1903, were analyzed in two different laboratories. The samples taken from the trial of June 18, 1903, were both analyzed in the same laboratory.

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ing each test, may be seen by a comparison of the various columns. Table VIII contains the analyses and heat values of the average samples of the coal, as well as the calculated heat value of the combustible matter. In this table are the percentages of ash found from the total coal fired on each day of the tests, as well as the heat values recalculated on the basis of the percentage of ash, as used in Table V. The percentage of ash found in the samples varied, as is evident from the table, between 4.6 and 10.5 per cent., while the tests themselves gave a minimum value of 6.1 per cent., the maximum value being 7.6 per cent. From two samples, taken on March 4, 1903, and analyzed in two different laboratories, the percentage of ash amounted to 9 and 5.88 per cent., respectively, while 6.96 per cent. was obtained from the test on the same day. Considering the amount of coal consumed, the latter value is probably the more accurate.

That the heat values of the first four tests could not vary from each other at the rate shown in Table VII, is proved by the agreement of the heat values of the combustible matter as well as by that of the parts of the refuse by weight (Table V). Besides, the close agreement of the values for the heat utilized in T.U. could—especially in the case of tests I and II—hardly be possible if the actual heat values so largely disagreed, as would be the case according to the average samples.

Even if the reason for recalculating the series of tests given in Table I (for which Table III contains the tabulation of the coal samples) is not so apparent, nevertheless, it seems to me quite opportune to refer to these conditions at this point; for, from what has been said, it follows that the determination of the heat value in the usual manner, viz: by the sample test, even if very carefully carried out, still forms a considerable source of error, which may be the cause of many discrepancies; and it is, perhaps, not unjustifiable to agitate this matter, in order to find out if a greater degree of certainty could be obtained.

The heat value of the combustible matter in coal is, without doubt, obtained with sufficient accuracy by the usual method

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in vogue. Therefore it would only be a question of determining the percentages of ash and moisture—which, of course, can vary considerably—with a greater degree of certainty, perhaps by using larger quantities for these examinations.

For considerable variations in the percentages of ash of the usual average samples, when compared to those values determined from the whole test, it seems to me—for comparative purposes—more accurate to base the comparison on the heat values as recalculated in the preceding manner, especially when sufficient care is taken in the determination of these values.

II. WAYS AND MEANS FOR THE ABATEMENT OF SMOKE DEVELOPMENT.

I now come to the second part of my paper, viz: the ways and means for the abatement of smoke.

Use of Special Fuels.

The simplest remedy for preventing the appearance of smoke is to use fuels that contain little or no volatile ingredients. In doing so we are not dependent upon the skill and attention of a fireman, and the ordinary grate, correctly proportioned, will be quite adequate.

Disregarding the fact that such fuels are not to be obtained in superabundant quantities, and therefore their extensive or compulsory use would, in general, greatly increase the cost, their use would in other respects be subject to certain restrictions. It is evident that fuel is less inclined to burn the fewer the volatile and easily ignited substances which are separated during combustion, therefore the less tendency there is to produce flame.

In cases where the variation in steam consumption is very marked and rapid, such fuels may fail, inasmuch as the heat development cannot follow with sufficient rapidity the increase in steam consumption, and therefore a constant pressure cannot be maintained, or for a decrease of the steam consumption the heat development cannot be checked rapidly enough.

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Both of these reasons would prevent the use of such fuels for many installations. Anthracite and coke, for example, burn without forming flame, and can only be used for very uniform work, or require, if the steam consumption is somewhat variable, sufficiently large water volumes, *i. e.*, amply large, and therefore moderately worked boilers. On account of the slow rate of combustion, such fuels, of course, require more grate area for the same boiler performance, which likewise leads to boilers that are large, expensive and occupy much space. Such fuels also require a stronger draft.

It is not, however, intended to argue that where these fuels can be employed or where the conditions for obtaining them are favorable they should not be used. In view of the fact that special fuels make it possible to obtain smokeless combustion, or at least combustion accompanied by light smoke, entirely independent of the attendance, they come into consideration principally for furnaces where the prevention of smoke *must* be assured; for example, installations situated in localities in which there is much traffic, or adjacent to public squares, etc., where intense smoke development would be found especially annoying and burdensome. It is impossible, however, for industry to restrict itself to the exclusive use of such fuels; for, without regard to the cost, coals rich in gas are generally, from an engineering and business point of view, far more valuable than those poor in gas.

There is yet to be observed that in many cases mixtures of different kinds of fuels, for example, anthracite screenings and coal, have proven advantageous. Experience with this kind of fuel for water-tube boilers has been very satisfactory.

The Employment of Special Methods of Attendance and Special Furnace Arrangements.

For coal rich in gas the question is—as fully discussed in part one—to regulate, as much as possible, the supply of air for the varying needs during stoking, if combustion is to be effected with slight smoke development and at the same time economically.

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As long as the coal consumption is moderate this can be attained in a satisfactory manner with the ordinary grate by careful working; the fuel being thrown only on the front part of the grate, after having pushed back the glowing coals. In this way the gas is slowly abstracted from the coal; the air required will vary but little, and it will not be difficult to obtain combustion accompanied by light smoke for a relatively slight excess of air. Smoke development can also be restricted by a uniform spreading of the coal in small quantities. But experience has shown that a larger excess of air will be necessary as the variation in the amount required is greater, and therefore the supply of air, if sufficient during the abstraction of gas, will subsequently be too large.

The moderate coal consumption required for the success of these methods of working naturally presupposes large installations, the existence of which no doubt would, in general, be in the interest of extreme utilization of fuel.

The greater the coal consumption the more difficult it becomes to work the ordinary grate with a moderate supply of air without developing considerable smoke, since more coal must be added in the various charges, or, if thrown near the front of the grate, the coal must be pushed back before gas abstraction has been completed. Therefore neither of these operations could be performed, without the production of smoke, should the air supply suddenly become insufficient.

The rate of combustion, especially when using coal rich in gas, must not fall below a certain limit, for without an excess of air the draft will be too light, the result being a smouldering fire and imperfect combustion.

As is well known, to assist the fireman in bringing about perfect combustion and at the same time to be less dependent on his skill, a great many contrivances have been invented. They can, in general, be divided into two main groups. One group endeavors to regulate the supply of air uniformly, either by continuous stoking or by causing first a slow abstraction of the gas from the coal, during which the volatile substances flow off over the glowing coals or through them. To this

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group belong furnaces with mechanical stokers, coal-dust furnaces, furnaces with inclined grates, basket-grate furnaces and similar arrangements. The other group retains the continuous stoking feature, and seeks to meet a possible variation in the air required by regulating the supply by the introduction of so-called secondary air in the burning gas stream immediately after the stoking and working of the fires.

I am well aware that many manufacturers are very suspicious of any special contrivances for restricting smoke development, and, as a matter of fact, this is fully justified in the case of a great many of those that have been proposed.

Apart from the failures caused by such faulty constructions, aversion to these contrivances has been frequently produced through the exorbitant claims for economy made by the over-zealous inventors. With respect to durability and safety in operating, much is yet desired. Frequently proper consideration is not given to the fact that the effectiveness of any arrangement is limited to certain conditions of manufacture; and, finally, disappointment has been caused by the fact that, even in well constructed furnaces, success is very much dependent upon the diligence of the fireman.

It is not my object to discuss fully these various constructions, though I would like to give the characteristics of the main groups mentioned.

For the large class of flue boilers having internal furnaces, for example, arrangements with upper-draft air supply occupy a prominent place and are not to be underrated. They are based on the well-known principle of providing the large quantity of air required for combustion immediately after stoking or working of the fires, by simultaneously introducing it, above and at the back of the grate, directly into the burning gas stream. For this, of course, the greatest variety in arrangement is possible. The air can be introduced at the front through the fire door or through special openings; it can be introduced at the sides of the grate, or the bridge wall can be used for this purpose (in which case it must be of a

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perforated construction); or, finally, it can be introduced in a special combustion chamber, built back of the bridge wall.

Even if these arrangements, without regard to their durability or adaptability, are not all equally effective (but will require a more or less large excess of air, according to the manner in which it is supplied), nevertheless the widely-accepted opinion that they *all* tend to produce light smoke does not quite agree, in the majority of cases, with the facts.

The effect of the upper-air supply can be easily demonstrated by leaving the fire door slightly open immediately after working the fires and as long as the main gas generation continues. I have caused this test to be frequently made during consecutive analyses of the waste gases and, when properly carried out, no impairment of the utilization was apparent, while the abatement of smoke was quite manifest.

Tables X and XI.

Data taken at the end of boiler.	Carbonic acid.	Oxygen.	Temperature of the gases.	Temperature of the air.	Draft, water column.	Waste heat loss.
Table X.	*per ct.*	*per ct.*	°C.	°C.	*mm.*	*per ct.*
Examination	8.8	10.	289	17	14.8	20.2
In charge of fireman instructor	12.8	6.1	292	16	9.5	14.
regular fireman after being instructed	11.7	7.3	276	14	8.4	14.5
First inspection	13.1	5.7	310	23	9.8	14.3
Second inspection	13.8	5.	314	31	8.	13.3
Table XI.						
Examination	8.	.....	339	20	11.	26.2
In charge of fireman instructor	11.	.....	326	20	4.	18.4
regular fireman after being instructed	11.4	.....	317	20	3.5	16.9
First inspection	10.9	8.4	335	28	3.5	16.2
Second inspection	12.4	6.6	349	13	6.	18.
Third inspection (lighter duty)	12.	7.3	239	18	2.5	12.

Relative to this, there is given in Table X an example of waste gas analysis, which clearly shows that, compared with the pernicious influence of the existing inadequate method of working (discussed in the foregoing), there can be no question of impairment of the utilization by supplying air through the fire door or in any other way, provided it is done effectively

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and at the proper time. For the conditions as they existed in the case which forms the basis of Table X, the average percentage of carbonic acid found at the end of the boiler was only 8.8 per cent., and the waste heat loss was about 20 per cent. At the same time the coal, which was of a quality strongly inclined to gas producing, gave off dense smoke after stoking and working of the fires. By expert working and regulating the supply of air in the manner indicated, an average proportion of carbonic acid of 12.8 per cent. and a decrease in the loss of the waste heat of about 14 per cent. could be obtained, the smoke making being quite moderate. From this it follows that it was not a question of smoke abatement alone, but that perfect combustion also set in for a moderate supply of air. After this a fireman-instructor was put in charge. Similar results, however, were also obtained from a test made with the regular firemen—after having been instructed, but entirely uninfluenced while in charge—as well as during subsequent inspections.

This remedy—leaving the furnace door slightly open to supply the necessary air—can, of course, be regarded only as a makeshift, for it demands of the fireman, especially if he has to look after several furnaces, very much attention. If the door remains open longer than necessary, an excess of air results and, consequently, an increase of the waste heat loss, as in the case of improper stoking.*

To obviate this a great many automatic devices have been invented, which at certain intervals of time admit the necessary air by means of a gradually-closing damper, correctly timed.

As a matter of fact, very satisfactory results can be obtained with such devices if correctly adjusted and expertly managed, as was demonstrated in a series of tests carried out by our association.†

* For this the fireman must be in a position to observe the top of the smokestack, an arrangement which, it may be said, should always be made.

† This association, for the purpose of clearing up certain questions (among which was that of the value of the upper draft), has carried out thorough tests under the most varying conditions. An installation was especially provided, for which an allowance of 6,000 marks was granted on the occasion of the anniversary of the German industry, and a report will be shortly forthcoming.

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In regard to the question of utilization in these furnaces, as compared with the ordinary grate, without doubt some inventors go too far. It is generally overlooked that even if on the one hand more perfect combustion is obtained, on the other hand the air supply, though the method of working remain the same, is somewhat increased, and with this the waste heat loss is greater; it follows from this that utilization can increase only when the former influences predominate.

Therefore it will not generally be a question of any *very* great differences; but, if actual differences have been established, they are generally not to be attributed to the principle of the furnace, but to a different method of working adopted by the fireman or to a more sparing supply of air (provided, of course, everything else has been correctly carried out as dedescribed in the foregoing).*

Furthermore, even if such devices make it easier for the fireman to work, with less smoke development than with the ordinary grate, the fact is self-evident that their effectiveness presupposes expert management. Their advantage, however, is in making it possible when using coal rich in gas to operate with a moderate excess of air, without incurring the danger of dense smoke making, which would be unavoidable in the case of failure of the secondary air supply under similar conditions. From an economical point of view such arrangements are not less dependent upon the fireman than the ordinary grate.

As soon as it becomes necessary to work with a greater supply of air to obtain perfect combustion more heat escapes—invisible, of course, but in any case not useful—through the stack, and the utilization suffers. After all, it cannot be maintained that such furnaces, without expert attendance, are less economical than the ordinary grate under the same conditions. In the numerous tests carried out by our Association we have, of course, encountered many such installations operating with bad utilization, where the average proportion of carbonic acid at the end of the boiler was 6 and 7 per cent., but still not more frequently than in the ordinary furnace

*See especially p. 656, last paragraph and the following.

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grate operated in the same manner. On the whole, it was shown that an average proportion of carbonic acid of 10 to 12 per cent. at the end of the boiler could be easily obtained in connection with satisfactory conditions of smoke making.

A further advantage of this type of furnace is the simple manner in which it can be combined with the ordinary grate and the good qualities of the latter be retained. They also permit a reasonably high coal consumption without losing any of their smoke-preventing qualities.

The main disadvantage of furnaces with an upper-air supply is that by incorrectly adjusting this draft more air is admitted than necessary, thus causing a loss in economy. This can be met by regular and expert supervision, as practiced by the Association for the Management of Furnaces and the Controlling of Smoke, of Hamburg.

As an example, I have chosen the results of an installation (to which might be added a series of similar ones) given in Table XI. The problem given here was that of four flue boilers equipped with Kowitzke furnaces. In connection with this it should be stated that the inspections were made unexpectedly, and the examination of the installation was made in the condition found at the time. The values clearly show that it was possible to work with quite a moderate excess of air and with satisfactory utilization. The combustion was satisfactory, producing a light smoke.

Any defects resulting from a change in the working force, different kinds of coal, or varying conditions of management, can be easily removed in this manner by frequent inspections, which the association undertakes to make once in every four months.

For the other large groups of furnaces having either continuous or interrupted stoking the process of combustion is not disturbed. In this case the excess of air naturally can be more readily diminished, without questioning the attainment of perfect combustion, so that generally from these arrangements higher values of utilization are to be expected than

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from the ordinary grate or from constructions that are only additions to the latter.

As a matter of fact, among these are to be found a series of thoroughly tried furnaces. I need only recall without entering into details the various arrangements of the Tenbrink furnace, inclined-grate furnaces, and similar constructions. In most of these furnaces is has been shown, especially for fuels strongly inclined to gas producing, that an upper supply of air into the burning gas stream is practicable. For the uniform combustion to be had in these furnaces such supply can be easily regulated.

The same applies to furnaces having mechanical stokers. Unfortunately, the useful constructions of this class—requiring infrequent repairs, and thus maintaining their good qualities—can, as a rule, be used only for coal of a uniform quality. If they could be made to burn, successfully, the ordinary run-of-mine coal, the question of their use would be of considerable importance, especially for large installations, where a smaller working force would be required.

These constructions are, in themselves, of course, more expensive than the ordinary grate, besides the additional attachments required. They also call for boilers partly of special construction, or require an increased expenditure for maintenance. The greater cost, it is true, is generally counterbalanced by a better utilization; but the paying qualities will always depend materially on the cost of the coal, and for this reason these arrangements have been largely introduced into localities where coal is expensive.

That a more perfect arrangement also requires to be properly handled and supervised, if it is to work continuously and in a reliable manner, is certainly a well-known fact, and this is of such importance that we cannot afford to neglect it.

It would be conducive to a wholesome progress if inventors would, first of all, relinquish the idea of improving the economy while trying to prevent smoke. At any rate more account should be taken of the actual conditions, which, I believe, I have to a certain extent set forth, and which indicate that the

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main losses are not to be sought in the smoke; consequently it should be considered satisfactory if we are able to bring about combustion accompanied by light smoke *at the same time preserving the economy.* If the economy is raised, which in many cases is not impossible, so much the better.

As long as coal is burned we shall never obtain a universal furnace that will meet the varied requirements, owing to the many properties of the fuels, to the peculiarities inherent in management, to different types of boilers, and to local conditions. In like manner, we need not anticipate new constructions obviating the making of smoke in a greater degree than heretofore.

Moreover, it must be particularly pointed out that the effectiveness of all furnace arrangements for smoke abatement are dependent upon the injurious effect on the boiler, and that even the best arrangements can fail if the rate of combustion (*i. e.*, the coal burnt per square meter of grate surface in a unit of time) exceeds certain limits.

Still, good results can yet be obtained, for certain condiditions, with various arrangements; but to exceed these conditions, even in the interest of economy, does not recommend itself.*

The question of attendance is of vital importance for all

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* Dense smoke formation is very often mainly attributed to overloading of the boiler. In reality there are not many installations in which this condition is such that it would not be possible to bring about satisfactory conditions through suitably designed furnaces and *expert* attendance. For example, coal containing much gas can be burnt on an ordinary grate, in a flue boiler, at the rate of 150 kg. per sq m. of grate surface per hour with a moderate excess of air, and little smoke, provided there is arrangement, properly regulated and handled, for an upper-air supply.

Increasing smoke formation with an increase of the load in a boiler installation can in most cases be traced to the fireman who keeps the grate well covered to satisfy the increased demand in steam consumption; in so doing he works with a smaller excess of air than would be the case if the load were lighter. If care is not taken after stoking to provide a proper supply of air, then dense smoke formation cannot be avoided.

Similar conditions obtain in the case of the draft. Installations in which the draft is so meager that the quantity of air required for perfect combustion could not be admitted are rarely found, while the condition of working with too much draft is nearly always the rule.

The increase in smoke formation with increasing load can often be traced to a decrease in the excess of air; therefore the bringing about of perfect combustion without proper expedients is impossible.

The case is by no means rare where, despite an increase in smoke formation, utilization improves with an increasing load by reason of a reduced excess in air. Such improvement occurs also when working, as is frequently the case, with a light load and full draft of the stack, *i. e.*, with the damper wide open.

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furnace arrangements. Upon this question I have dwelt at length for furnaces having a secondary air supply.

Education and Supervision of the Firemen.

Correct training and, more particularly expert supervision and control of the fireman, is of fundamental importance in the whole question of management of furnaces.

The Association for the Management of Furnaces and Controlling of Smoke, of Hamburg, takes hold of this question in the following manner:

"After all the conditions have been thoroughly examined by an engineer, a fireman instructor, through his own efforts and example, shows the fireman on the ground what may be obtained for the special conditions. Based upon the results thus obtained, the firemen are then placed under the direction of a fireman instructor until they have thoroughly mastered the various manipulations. After this the examination is repeated by the engineer, and the results noted, the progress made being shown the firemen, and their knowledge of the correct management of a furnace thus increased. At the same time, the association seeks to impress upon the managers or others in charge the need of supervision and systematic control to prevent relapse into former bad practice.

"In this, we believe, we are on the right path and, in any event, attain more in this manner than through fireman schools, fireman examinations," etc.

The establishment of schools for firemen was also considered by our association in Hamburg; but I am of the opinion that the effect of such schools upon the management of furnaces will not be as marked as is generally expected. The whole question is not so much one of knowledge as of a certain, easily-obtainable, skill; *and more particularly is it a matter of reliability and constant attention to the main points at issue.*

Therefore, owing to the shortcomings of mankind, without a constant and expert supervision, permanent success will never

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be achieved. Such supervision would continually tend toward, and produce rules for, correct working, and it would also take due cognizance of any variation in the conditions.

Similar instructions repeatedly given in specially selected installations are of course much more simple and less tedious, full regard being had to the peculiarities of management in each case, than instructing individually on the ground. However, it is an ever-recurring experience, with the varying conditions exhibited in this field, that firemen, in most cases, can only be convinced by showing them in their own installations with its individual peculiarities, that improved working is possible.

It is likely that courses of instruction in this branch would meet with more success if they were given to those in charge of firemen. For in these very circles it is of the utmost importance that the understanding of the procedure for the management of furnaces be better than it is at present, as success depends very materially on the expert, and, if necessary, determined manner of supervision during working. Influence in this direction is also to be obtained in a thoroughly satisfactory way along with the method adopted by our association for the education of firemen.

In regard to the duty of fireman instructors, I wish to say further that fireman instructors have now been in vogue elsewhere for some length of time. But my observation of this institution has convinced me that if anything really substantial is to be done, the efforts of the fireman instructor must be guided by a trained engineer. As a rule, the fireman instructor himself can only be one who carries out orders. In the case of the judgment of furnaces—simpler installations excepted—problems are presented which demand for their solution more knowledge and ability than can be expected of a fireman instructor who comes from the working classes.

I, therefore, proceeded in such a manner as to have the association engage, first, one scientifically-trained engineer and later on another; both were familiar with the management of boilers, and besides, they were required, for some

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time, to work as firemen in various installations. Each of these engineers is given a fireman instructor, and the necessary examinations, instructions, etc., are carried out by them together. I believe the results thus far obtained have justified this method.

III. ECONOMY GAINED IN SUPERVISING THE MANAGEMENT OF FURNACES.

It is my wish to mention the importance—not to be underrated—of the examination and supervision of installations when carried out in the manner indicated; and I would also especially refer, from the view point of economy, to the analysis of the waste gases.

From the first part of my subject it is clear that the observation of the top of the smoke stack alone in no way suffices to form an adequate judgment of the working of a furnace, but is of real value only when combined with a correct analysis of the waste gases.

Of course these analyses are little comprehended at first. It is recalled that the factor of evaporation is something more tangible, and that the knowledge of the same for judging the conditions in a practical way is sufficient. But this is only correct up to a certain point; for, apart from the fact that reliable determination of the factor of evaporation is comparatively circumstantial, and from the further fact that many sources of errors existing in establishments are unknown to us, its value is always to be regarded only as a relative one. Its importance is by no means to be misjudged, but taken alone, without simultaneous analysis of the waste gases, it never shows whether the fuel in an installation has been actually utilized, as might be reasonably expected; while this always follows from waste gas analyses. These analyses not only show, in a simple manner, about how much is yet to be gained under the conditions existing, but with their assistance it is also comparatively easy to establish whether the cause of inefficient utilization results from the method of working adopted by the fireman, badly proportioned grate surfaces, porous brick-

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work, poorly cleaned heating surfaces, errors in the lead of the gases, or the conditions of the draft, etc.

As a matter of fact, the waste gas analysis when correctly carried out is an *extremely valuable means of supervision* for actual practice. They hold at least the same relation to the furnace as an indicator test for a steam engine.

As to the significance, however, of the waste heat loss for the utilization, which can only be found in this manner, we are, in general, not yet well informed.

The opinion often prevails, even in well arranged establishments, that so long as there is no smoke made, everything else is satisfactory. In this we either forget, or else we are not clear upon the subject, that a much greater excess of air might exist (even in the case of a good arrangement) than would be necessary to obtain, when working with efficient attendance, light smoke. Hence, a regular supervision of the management of furnaces by means of waste gas analysis is in any case very valuable, for only thereby can it be established whether the fire in relation to the smoke and in relation to the utilization of the fuel is properly managed.

IV. DOMESTIC FURNACES.

As already mentioned in the introduction, in many cities, during the winter months, responsibility for the smoke nuisance rests as much with the domestic furnace as with furnaces of industrial establishments.

A partial relief can of course be obtained in this field by the use of fuels making light smoke, such, for example, as lignite briquettes, which are much used in Berlin.

Coke and anthracite are also well adapted for domestic purposes. But, despite the economical arrangements to be had for the latter fuels in the self-feeding stoves, their use, as well as those of briquettes, is already limited, owing to the fact that in many places they are considerably more expensive than coal and an increased demand would raise their cost. It would also be impossible to obtain them everywhere in the requisite quantities for general use.

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By improving the construction of the stoves something, to be sure, might yet be obtained; but, precisely as in the case of the large furnaces, the effectiveness of these small stoves, even for the best constructions, depends on expert attendance. This is, however, for the class of employés in question, not to be expected in an adequate degree, and a successful influence in this direction would be impossible.

Without doubt more can be obtained from central heating stations, which are constantly becoming more numerous, despite the fact that detached furnaces burning coal are inefficient as well as inconvenient. In the smaller of these central stations coke furnaces are generally used on account of the low percentage of gas contained in this fuel and the attendant simple and safe working conditions, with no smoke nuisance. In the case of the larger of these central stations—the heat generators being generally steam boilers, with large furnaces requiring constant attendance—they can be treated in the same manner as the furnaces of regular industrial plants.

Perhaps the present competition between the gas engine and steam engine may result in producing a cheaper gas for general use. This would make a more extended use of gas possible for cooking and heating, forming, of course, the most effectual relief.

To briefly recapitulate the ways and means for the abatement of smoke in industrial furnaces we have the following points:

1. *Education and supervision of the firemen.*

2. *Improving the furnaces with respect to construction and management.*

3. In case nothing can be attained in any other manner, or if we want to be absolutely certain, then *the use of fuels producing light smoke.*

None of these ways, taken alone, is wholly satisfactory, but by taking them all together, and perseveringly pursuing them, without doubt progress can be made.

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At this point, gentlemen, I have, practically, concluded my address, but I wish to add a few general comments.

My deductions may have shown that, as respects smoke abatement, nothing satisfactory is to be obtained by way of a smoke prohibition law, because the conditions for that are entirely too complicated. If, furthermore, industrial interests should not only object to the compulsory use of certain kinds of fuels, but also to a general law for the employment of special furnace arrangements, then, in my opinion, owing to the conditions which prevail in this field, such objections would be, in a great measure, justifiable. The question is not only one of providing such special arrangements, *but they must also be adapted to the existing conditions, and, further, must be correctly managed and intelligently supervised.* The latter point—skilled and intelligent supervision—is of the greatest importance for the entire success of these arrangements. Therefore, the authorities would also have to adopt the needful precautionary measures in this direction if such laws are to be of any benefit. But considering such municipal measures, and what has been, and might yet be accomplished by them, I believe that an easier and surer solution would be obtained through concerted and harmonious action on the part of the manufacturers interested; that is, a policy of mutual help, such as pursued by the Hamburg association. The management of these combined interests should be assured of an opportunity to exert educational pressure, and also receive the confidence and support of the city authorities, such as is given to the societies for boiler inspection. All this would be most conducive to the aims of such a union among the manufacturers; but it is evident that, besides, the management should possess the power to act in an advisory capacity in the case of any new installations made in establishments which come under its supervision.

For industry, the way here outlined has the absolute advantage, that (except in some cases, where the use of special arrangements cannot possibly be avoided) full attention is given to obtain extreme utilization of the fuels, and the owner is

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guaranteed that all conditions are duly considered. Moreover, the owner may command at any time expert advice on all questions relating to the management of furnaces; his plant in regard to economy will be regularly controlled, and his employés will be constantly guided and instructed. But such matters as these (and they are not matters for the municipal authorities either to consider or settle) could all be readily solved by the officers of the association, owing to their close and constant contact with the various establishments and knowledge of peculiarities of management, etc. It would not be difficult for them to discover the means needed to restrict smoke development, without exposing the owner to the risk of costly experiments.

With this I shall conclude my remarks. Owing to the vital nature of this problem, the industrial world is much interested in the proper method of procedure, and in my discussion I have endeavored to describe the start which our association has made, and give an idea of the work already accomplished along the lines pursued. My statements will have achieved their end if, together with the association's record up to the present, some profit to local interests results, and an incentive is given elsewhere to further progress in dealing with this important question for the general welfare.

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THE U. S. S. *GALVESTON.*

GENERAL DESCRIPTION—STANDARDIZATION, OFFICIAL AND ENDURANCE TRIALS.

The *Galveston* (*Cruiser No. 17*) is one of six sheathed, protected cruisers, known as the *Denver* type, authorized by Congress in 1899.

The contract for the *Galveston* was awarded to the William R. Trigg Co., of Richmond, Va., December 14, 1899, who were to deliver her completed in thirty months, or in June, 1902. Contract price, $1,027,000. Owing to various delays, due to non-delivery of material, the time for completion was extended to September 15, 1903. On December 24, 1902, the W. R. Trigg Co. failed, and the contract was declared forfeited May 14, 1903. A board of naval officers was appointed to make a complete inventory of all work done or commenced and of all material on hand, for the purpose of completing the vessel.

The Board found the hull 66 per cent. and the machinery 77.25 per cent. completed.

In September, 1903, the vessel was removed to the Navy Yard, Norfolk, Va., where she was completed by the Government, and commissioned February 15, 1905.

As complete descriptions of this class of ships have appeared at various times in the JOURNAL, it will be only necessary to repeat here the principal dimensions. For more extended data, the reader is referred to Vol. XV, page 1110 (Trial of U. S. S. *Cleveland*), Vol. XVI, page 1 (U. S. S. *Tacoma*), page 67 (U. S. S. *Denver*, page 227 (U. S. S. *Des Moines*), and Vol. XVII, page 104 (U. S. S. *Chattanooga*).

HULL.

The hull is constructed of mild steel, the outer plating being composed generally of 15-pound plating. There are one hundred transverse frames, spaced 3 feet apart, and formed of

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Z-bars (frames 18 to 36), 6 inches by 3½ inches by 3½ inches by 15½ pounds, except forward and aft (frames 1 to 17 and 87 to 100), which are 4 inches by 3½ inches by 9.1 pounds angle bars. The outside plating is sheathed with Georgia pine, 4 inches thick, extending to about 42 inches above the water-line amidships, shearing up forward to 69 inches, and aft to 51 inches.

Length L. W., feet	292
over all, feet and inches	308- 9
Beam molded, feet and inches	43- 3¼
extreme, feet	44
Mean draught (with 467 tons coal and two-thirds stores), feet and inches	15- 9
Corresponding displacement, tons	3,200
Area of midship section, square feet	615.2
load-water plane, square feet	9,370
Tons per inch of immersion	22.3

ARMAMENT.

Main battery	10 5-inch R. F. guns, 8 6-pdrs.
Secondary battery	2 1-pdrs., 4 Colt's automatic.
Mean height of axis of 5-inch guns:	
Main deck, above load-water line, feet and inches	21- 5
Gun deck, above load-water line, feet and inches	12- 4

MACHINERY.

There are two triple-expansion, four-cylinder engines, of the vertical, inverted type, placed abreast of each other in separate watertight compartments, and designed to develop a total of 4,500 I.H.P. at about 172 revolutions. The engines are right-and-left, turn inboard, and have the following sequence of cylinders, beginning forward: Forward low pressure, high pressure, intermediate pressure and after low pressure. There are two main condensers with a cooling surface of 3,004 square feet each.

Steam is supplied by six water-tube boilers of the Babcock & Wilcox *Alert* type.

ENGINES.

Number of engines	2
cylinders, each engine	4
Stroke (common), inches	30
Diameter H.P. cylinder, inches (originally 18 inches)	18⅛
I.P. cylinder, inches	29
L.P. (2) cylinders, inches	35½

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Cylinder clearances:	*Stbd. engine.* *Top.*	*Stbd. engine.* *Bottom.*	*Port engine.* *Top.*	*Port engine.* *Bottom.*
H.P. cylinder, inch	$\frac{7}{32}$	$\frac{7}{16}$	$\frac{5}{32}$	$\frac{7}{16}$
I.P. cylinder, inch	$\frac{9}{32}$	$\frac{7}{16}$	$\frac{9}{32}$	$\frac{25}{64}$
Forward L.P. cylinder, inch	$\frac{15}{32}$	$\frac{1}{2}$	$\frac{13}{32}$	$\frac{1}{2}$
Aft L.P. cylinder, inch	$\frac{21}{64}$	$\frac{15}{32}$	$\frac{1}{2}$	$\frac{19}{32}$

Valves, H.P. (one piston for each cylinder), diameter, inches 11

I.P. (two for each cylinder), diameter, inches 11

L.P. (one double-ported slide for each cylinder), inches, length, $25\frac{1}{4}$; width, $39\frac{1}{2}$; port, length 37

Table I.—DETAILS OF VALVE SETTINGS.

	H.P.		I.P.		L.P.	
Side on which steam is taken	Inside.		Outside.		Outside.	
Travel of valve, inches	4		4		4	
	Top.	*Bottom.*	*Top.*	*Bottom.*	*Top.*	*Bottom.*
Width of port, inches	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}\times37$	$1\frac{1}{2}\times37$
Steam lap, inch	$\frac{29}{32}$	$\frac{27}{32}$	$\frac{29}{32}$	$\frac{27}{32}$	$\frac{15}{16}$	$\frac{13}{16}$
Exhaust lap, inch	$-\frac{5}{16}$	$-\frac{1}{16}$	$-\frac{5}{16}$	$-\frac{1}{16}$	...	$+\frac{1}{4}$
Angular advance, degrees	$35\frac{1}{2}$	$35\frac{1}{2}$	$35\frac{1}{2}$	$35\frac{1}{2}$	40	40
Steam lead, angular, degrees	8	10	8	10	12	16
linear, inch	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{11}{32}$	$\frac{15}{32}$
Cut-off, in inches	$23\frac{3}{8}$	$21\frac{3}{16}$	$23\frac{3}{8}$	$21\frac{3}{16}$	$22\frac{1}{16}$	$20\frac{3}{16}$
decimal of stroke	0.78	0.705	0.78	0.705	0.735	0.672
Exhaust release, in inches	$3\frac{7}{16}$	$3\frac{5}{8}$	$3\frac{7}{16}$	$3\frac{5}{8}$	$2\frac{13}{16}$	$2\frac{7}{8}$
decimal of stroke	0.114	0.1205	0.114	0.1205	0.093	0.095
Compression, in inches	$1\frac{7}{8}$	$1\frac{15}{16}$	$1\frac{7}{8}$	$1\frac{15}{16}$	$4\frac{3}{16}$	$3\frac{1}{8}$
decimal of stroke	0.062	0.064	0.062	0.064	0.139	0.129
Steam opening, inches	$1\frac{3}{32}$	$1\frac{5}{32}$	$1\frac{3}{32}$	$1\frac{5}{36}$	$1\frac{1}{16}\times2$	$1\frac{3}{16}\times2$
Exhaust opening	Full port.		Full port.		Full port.	
Velocity of steam, in feet per second, (860 ft., 172 revs., per minute)	128.6	121.6	166.9	157.9	189.3	169.4
Velocity of exhaust, in feet per sec., (860 ft., 172 revs., per minute)	93.82	93.82	121.7	121.7	134.12	134.12
Velocity of exhaust to condenser	115.6 feet per second, $12\frac{1}{2}$ nozzle.					

PROPELLERS.

Starboard ..... Right hand.

Port ..... Left hand.

Number of blades 3

Diameter, feet and inches 10- 6

Pitch, feet and inches (as set on trial) ..... 11- $8\frac{1}{2}$

adjustable, feet and inches ..... 11- 2 to 12- 5

feet and inches (as designed) 11- 9

Helicoidal area, each screw, square feet 34

Projected area, each screw, square feet 29

Disc area, square feet 86.6

Pitch ratio, $\frac{P}{D}$ ..... 1 : 115

BOILERS.

Type ..... Babcock & Wilcox, marine.

Number of boilers 6

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Length, feet and inches	9–10½
Width, feet and inches	9– 0½
Height to center of steam drum, feet and inches	10– 3
Diameter of steam drum, inside, inches	42
Length of steam drum, feet and inches	9– 6⅝
Thickness of steam drum, inch	$\frac{21}{32}$
Number of furnaces per boiler	1
furnace doors per boiler	3
Length of grates, feet and inches	6– 6
Width of grates, feet and inches	7– 7
Diameter of tubes, feet and inches	2 and 4
Thickness of 2-inch tubes	No. 8 B.W.G.
4-inch tubes	No. 6 B.W.G.
Number of 2-inch tubes per boiler, 8 feet 11 inches long	406
4-inch tubes per boiler, 8 feet 11 inches long	17
Number of circulating tubes per boiler, 7 feet 3 inches long	15
Total number of 2-inch tubes	2,436
4-inch tubes	192
Heating surface per boiler, square feet	2,200
Total heating surface, square feet	13,200
Grate surface per boiler, square feet	50
Total grate surface, square feet	300
Ratio heating surface to grate surface	44 : 1
Tube-heating surface per boiler, square feet	2,172
Total tube-heating surface, square feet	13,032
Plate-heating surface per boiler, drum, square feet	28
Total plate-heating surface, drums, square feet	168
Number side boxes, 8 feet 3 inches × 5⅝ × 5⅝ O.S. by ⅝-inch thick per boiler	8
Number side boxes, 8 feet 3 inches × 5⅝ × 5⅝ O.S. by ⅝-inch thick per boiler, total	48
Number corner boxes, front, per boiler, $38\frac{7}{16}$-inch × 5¾-inch I.S. × ⅝-inch thick	2
Number corner boxes, front, total	12
Number corner boxes, rear, per boiler, $39\frac{13}{16}$ inch × 5¾ × 5¾ inches I.S. × ⅝-inch thick	2
Number corner boxes, rear, total	12
Number front cross boxes per boiler, 7 feet 6¼ inches × 5¾ inches × 5¾ inches I.S. × ⅝-inch thick	1
Number front cross boxes, total	6
Number of smoke pipes	2
Diameter of smoke pipes for two forward boilers, feet and inches	4– 4
Diameter of smoke pipes for four after boilers, feet	6
Total area of smoke pipes, square feet	43
Height above grate bars, feet	70
Ratio smoke-pipe area to grate surface	1 : 6.98
Pressure for which designed, pounds per square inch	275

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Table II.—AUXILIARY MACHINERY U. S. S. *GALVESTON.*

Purpose for which used.	No.	Type.	Make.	Diameter steam cylinder.	Diameter water cylinder.	Stroke.	Weight of each.	Remarks.
				inches.	*inches.*	*in.*	*pounds.*	
Main air pump	2	Single-acting.	Trigg.	.....	20	10	.....	Worked by beam, forward L.P. crosshead.
Comb. aux. air and circulating pump	2	Double-acting.	M. T. Davidson.	6	Air 7 Water 10	10	1,981	For auxiliary condensers.
Auxiliary condenser	2	.....	.....	.....	.....	...	2,402	400 sq. ft. cooling surface, each.
Main circulat'g pump	2	Centrifugal.	Trigg.	(2) 6	Runner 36	6	.....	Suction and discharge diameter, 12 inches.
Feed pumps	4	Vert. simplex.	Blake.	10	7	12	2,070	Outside-packed plunger.
Fire and bilge pumps	4	Vert. simplex.	Blake.	7	7	12	1,015	2 in engine rooms, 2 in fire rooms.
Water-service pumps	2	Vert. simplex.	Blake.	7	7	12	1,057	1 each engine room.
Hotwell pumps	2	Vert. simplex.	Blake.	7	7	12	1,045	1 each engine room.
Shaft bilge pump	2	Single-acting plunger.	Trigg.	.....	3	8	202	On forward end of shaft.
Evap. feed pump	1	Vert. simplex.	Blake.	4½	3½	6	.....	
Forced-draft blowers	3	Fan.	B. F. Sturtevant.	(2) 5	Fan 60	4	2,467	
Dynamo and engine	4	Compound.	W. D. Forbes.	7½–13	.....	7½	8,412	Direct connected to Bullock 25 kw., 80 volt generators.
Capstan and anchor engine	1	Double.	Hyde.	(2) 9	.....	9	18,900	
Steering engine	1	Double.	Williamson.	8	.....	10	13,005	
Distillers	3	.....	Bureau design.	.....	.....	...	556	2,500 gallons each.
Ash-hoist engine	2	.....	Williamson.	4½	.....	4½	.....	
Evaporators	2	.....	M. T. Davidson.	.....	.....	...	6,297	5,000 gallons each.
Ice machine	1	.....	Allen dense air.	7	.....	10	5,950	One ton capacity.

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MACHINERY WEIGHTS IN DETAIL.

Actual weights of the various parts of the finished machinery, as built and installed, are as follows:

Group.		*Tons.*
I.	Main engine cylinders, receiver pipes, etc	26.55
II.	Shafting	20.15
III.	Main engine framing and bearings	23.52
IV.	Main engine reciprocating parts	7.23
V.	Main engine valve gear	6.53
VI.	Main condensers	12.91
VII.	Main air and circulating pumps	5.76
VIII.	Propellers	5.32
IX.	Boilers	97.97
X.	Boiler fittings	37.70
XI.	Smoke pipes and uptakes	24.70
XII.	Steam and exhaust pipes and valves	15.05
XIII.	Suction and discharge pipes and valves	18.19
XIV.	Lagging and clothing	8.15
XV.	Flooring, gratings, etc	15.26
XVI.	Auxiliaries	18.43
XVII.	Fittings and gear	15.77
XVIII.	Water in boilers and machinery	37.10
XIX.	Stores, tools and spare parts	19.20
XX.	Miscellaneous machinery, heating, distilling and refrigerating plants	28.76
XXII.	Connections under steam engineering to other miscellaneous machinery	3.78
	Total	448.03

STANDARDIZATION OF SCREWS.

Progressive runs were made over a measured mile for the purpose of standardizing the screws and thus determining the number of revolutions to be made by the engines in order to give the speed for four hours at sea, required by the contract. The course off Provincetown, Mass., was selected for this purpose, and the runs with and against the tide made at the speed shown on Table III, from which the curves on plates 1 and 2 were plotted.

Table IV gives the general performance data as recorded or calculated.

The runs over the measured mile were made in a smooth sea, with calms to light southerly airs during the first eight runs and a gentle southwest breeze during the remainder.

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Table III.—U. S. S. *GALVESTON.*

Data from Progressive Runs over Measured-Mile Course off Princetown, Mass., May 29, 1905.

Number and direction of run.	Time on course.	Revolutions of main engines. To make one knot. Starboard.	Revolutions of main engines. To make one knot. Port.	Revolutions of main engines. Per minute. Starboard.	Revolutions of main engines. Per minute. Port.	Revolutions of main engines. Per minute. Mean.	Speed in knots per hour.	Indicated horsepower. Starboard engine. H.P. cylinder.	Starboard engine. I.P. cylinder.	Starboard engine. F.L.P. cylinder.	Starboard engine. A.L.P. cylinder.	Starboard engine. Total.	Port engine. H.P. cylinder.	Port engine. I.P. cylinder.	Port engine. F.L.P. cylinder.	Port engine. A.L.P. cylinder.	Port engine. Total.	Total both main engines.	Auxiliaries in operation (estimated).	Total all machinery.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
	m. s.																			
1-S	9-57.8	610.9	588	61.3	59	60.65	6.022	34.84	32.41	10.94	10.94	89.13	32.42	29.16	8.78	9.66	80.02	169.15	25.48	194.63
2-N	8-15.91	625.6	611.6	75.6	74	74.8	7.26	63.31	56.41	16.87	12.37	148.96	54.68	43.88	14.31	17.07	129.94	274.9	23.46	298.36
3-S	6-33.1	591.6	586.2	90.1	89.3	89.7	9.142	96.1	96.17	28.82	19.44	240.53	91.36	75.02	25.91	29.89	222.18	462.71	25.58	488.29
4-N	6-04.74	638.1	635.5	104.9	104.5	104.7	9.871	148.06	145 67	49.16	35.9	378.79	138.59	129.1	48.2	42.76	358.65	737.44	29.31	766.75
5-S	4-59.44	613.5	608.1	122.9	121.9	122.4	12.024	224.69	242.93	90.05	74.97	632.64	207.85	210.84	76.17	75.27	570.13	1,202.77	34.11	1,236.88
6-N	4-41.48	649.4	653.5	138.4	139.3	138.8	12.789	320.94	324.18	129.21	111.19	885.52	300.85	309.77	124.35	129.53	864.5	1,750.02	36.8	1,786.82
7-S	4-21.12	626.5	629	143.9	144.5	144.2	13.788	365.31	366.92	156.29	156.29	1,044.81	355.88	357.03	150.49	155.86	1,019.26	2,064.07	47.43	2,111.5
8-N	4-20.39	666.1	667.1	153.5	153.7	153.6	13.825	503.11	474.09	183.84	189.55	1,350.59	419.3	425.33	177.22	182.94	1,204.79	2,555.38	48.65	2,604.03
9-S	3-54.38	639.8	639.2	163.8	163.6	163.7	15.356	550.82	586.84	231.51	220.55	1,589.72	548.6	557.83	237.31	237.31	1,581.05	3,170.77	71.93	3,242.7
10-N	3-55.16	677.4	691	172.8	176.3	174.6	15.306	641.66	713.02	300.79	278.94	1,934.41	647.96	688.25	305.05	308.2	1,949.46	3,883.87	70.79	3,954.66
11-S	3-32.95	665.1	666.6	187.4	187.8	187.6	16.901	809.46	926.06	405.66	414.04	2,555.22	800.53	909.5	408.77	412.13	2,530.93	5,086.15	65.25	5,151.4
12-N	3-44.81	694.8	717.9	185.4	191.6	188.5	16.014	772.74	911.6	402.72	426.17	2,513.23	675.17	946.83	560.14	534.49	2,716.63	5,229.86	115.54	5,345.4
13-S	3-35.63	660.5	663.1	183.8	184.5	184.1	16.698	612.86	876.5	512.73	408.82	2,410.91	594.23	927.39	528.41	511.25	2,363.28	4,774.19	97.96	4,872.15

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Table IV.—U. S. S. *GALVESTON*.

Data from Progressive Runs over Measured-Mile Course off Provincetown, Mass., May 29, 1905.

Number of run.	Time at beginning of run (approximate).	Which engine.	Steam pressures. In boilers above atmosphere.	Steam pressures. At engine above atmosphere.	Steam pressures. H.P. steam chest, absolute.	Steam pressures. In first receiver, absolute.	Steam pressures. In second receiver, absolute.	Mean effective pressures. H.P. cylinder.	Mean effective pressures. I.P. cylinder.	Mean effective pressures. Forward L.P. cylinder.	Mean effective pressures. After L.P. cylinder.	Mean effective pressures. Referred to low pressure.	Vacuum, inches.	Cut off in cylinders from beginning. H.P.	Cut off in cylinders from beginning. I.P.	Cut off in cylinders from beginning. Forward L.P.	Cut off in cylinders from beginning. After L.P.	Temperatures. Outside air.	Temperatures. Engine-room working platform.	Temperatures. Fire-room working level.	Temperatures. Injection.	Temperatures. Discharge.	Temperatures. Hotwell.	Temperatures. Feed.	Revolutions or double strokes. Circulating pump.	Revolutions or double strokes. Main feed pumps.	Revolutions or double strokes. Hotwell pump.	Revolutions or double strokes. Water-service pump.	Revolutions or double strokes. Bilge pump.	Revolutions or double strokes. Aux. cond'r. pump.	Revolutions or double strokes. Fire-room blowers.	Air pressure, inches water.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33
1-s	2·18	St'b'd	208	207	35	13	6	15	5.35	1.2	1.2	4.887	27	Cut off at .742.	Cut off at .742.	Cut off at .703.	Cut off at .703.	75	89	107	50	69	100	210	110	14	12	57	50	34	...	...
		Port	208	207.5	33	13.5	4	14.5	5	1	1.1	4.557	27						93		50	61	82	200	160	9	11		38		...	...
2-n	2·45	St'b'd	208	207	47	17	6.5	22 1	7.55	1.5	1.1	6.621	27.2					75	89	110	50	70	112	185	110	12	9	58	30	35	...	...
		Port	208	207.5	42	15	5	19.5	6	1.3	1.55	5.901	27						93		50	62	80	204	152	9	12		48		...	...
3-s	3·12	St'b'd	210	207	60	21	7	28.15	10.8	2.15	1.45	8.971	27.5					74	90	108	50	70	112	200	120	12	11	56	32	32	...	...
		Port	210	208.5	58	20	6	27	8.5	1.95	2.25	8.361	27.5						93		50	64	74	194	156	12	24		49		...	...
4-n	3·33	St'b'd	215	207	75	27	9	37.25	14.05	3.15	2.3	12.135	28					74	90	107	50	80	112	205	120	16	12	57	30	34	...	...
		Port	214	209.5	74	26	7	35	12.5	3.1	2.75	11.534	27.5						93		50	67	72	186	156	15	26		48		...	...
5-s	3·55	St'b'd	245	249	92	37	11	48.25	20	4.925	4.1	17.302	28					73	90	110	50	80	92	215	120	20	12	63	42	36	...	...
		Port	246	227.5	96	33	9	45	17.5	4.2	4.15	15.718	27.5						90		50	70	67	180	168	16	24		56		...	...
6-n	4·13	St'b'd	255	228	125	40	13	61.2	23.7	6.275	5.4	21.502	28					73	90	103	50	87	92	215	130	20	14	64	38	36	170	...
		Port	255	232.5	122	42	12.5	57	22.5	6	6.25	20.856	27.5						90		50	78	70	180	168	19	25		36		195	...
																															160	
7-s	4·31	St'b'd	270	249	157	55	15	67	25.8	7.3	7.3	24.4	28					72	87	99	50	85	90	210	130	39	18	64	36	40	170	...
		Port	270	261.5	137	49	14.5	65	25	7	7.25	23.708	27.5						90		50	78	71	194	159	18	28		60		205	...
																															155	
8-n	4·49	St'b'd	260	249	167	57	22	86.5	31.25	8.05	8.3	29.57	28					72	90	92	50	85	90	190	135	28	34	64	40	40	170	...
		Port	260	241.5	151	53	15.5	72	28	7 75	8	26.344	27.5						90		50	78	71	194	152	30	23		52		215	...
																															150	
9-s	5·04	St'b'd	265	249	190	65	23	88.75	36.25	9.5	9.05	32.615	28					72	84	96	50	85	90	180	145	50	40	64	50	40	180	...
		Port	265	257.5	185	64	19.5	88.5	34.5	9.75	9.75	32.48	27.5						90		49	84	69	170	152	34	48		54		220	.2
																															155	
10-n	5·26	St'b'd	275	259	220	75	25	98	41.75	11.7	10.85	37.62	27.5					72	90	92	50	95	95	185	145	44	40	64	50	36	356	...
		Port	275	250.5	215	78	25	97	39.5	11.63	11.75	37.162	27						90		49	85	74	169	162	40	32		80		340	.9
																															370	
11-s	5·40	St'b'd	280	256	262	100	30	114	50	14.55	14.85	45.82	27					71	90	87	50	110	108	180	160	40	50	62	50	36	450	...
		Port	280	251.5	237	97	30.5	112.5	49	14.63	14.75	45.29	26.5						90		49	100	81	184	172	33	36		46		395	1.6
																															375	
12-n	5·52	St'b'd	280	254	260	98	30	110	49.75	14.60	15.45	45.555	27.5					71	90	93	50	95	90	180	180	62	50	61	50	36	489	...
		Port	280	246.5	243	105	37	93	50	19.65	18.75	47.65	26.5						90		49	91	86	192	196	60	26		34		471	2
																															433	
13-s	6·06	St'b'd	260	259	265	102	33	88	48.25	18.75	14.95	44.08	27.5					71	86	90	50	95	90	180	180	60	50	60	50	40	528	...
		Port	260	234.5	235	93	36	85	40	19.25	18.63	43.05	26.5						90		48	94	84	170	180	56	40		52		507	2
																															421	

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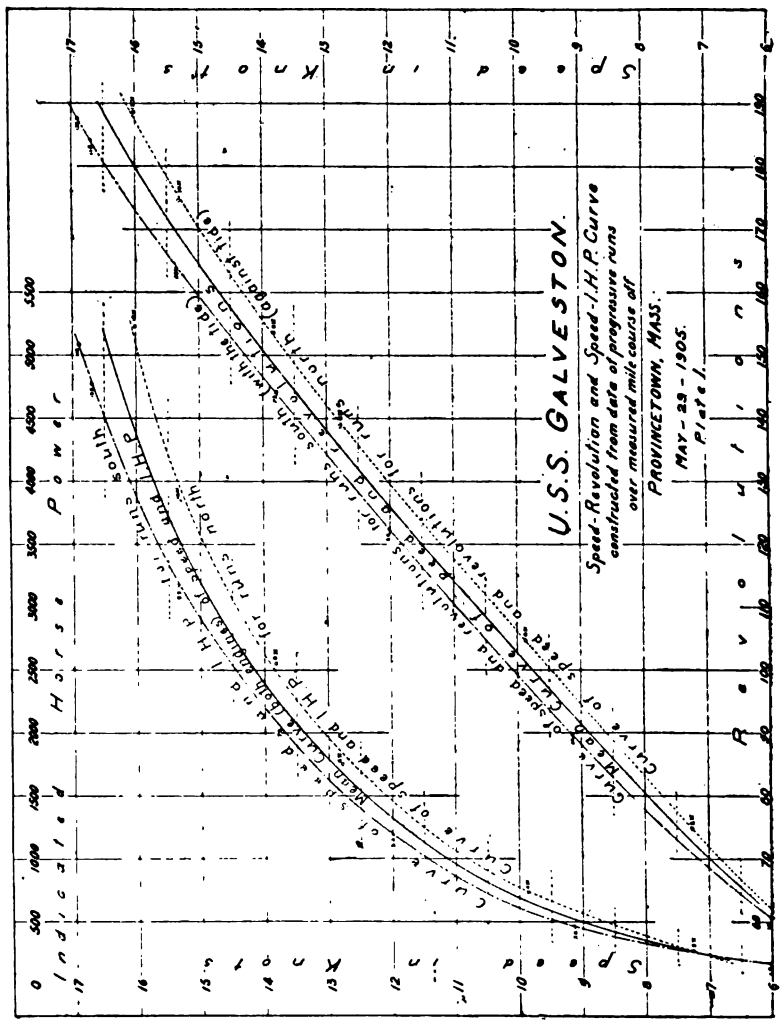

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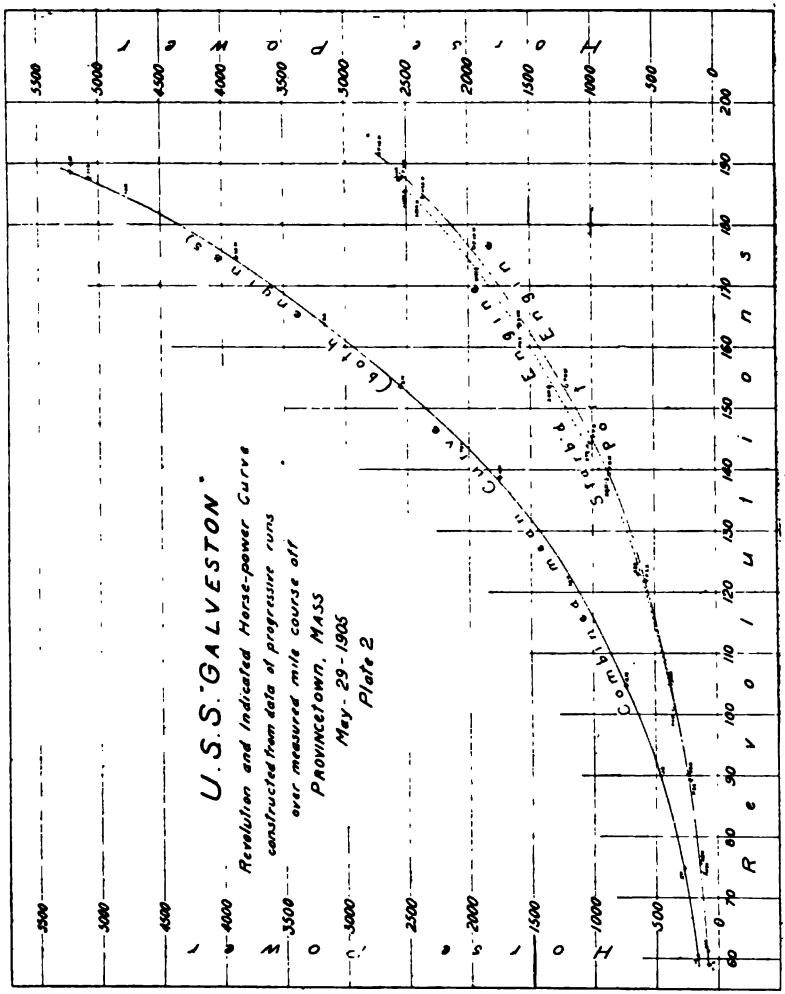
U.S.S. "GALVESTON"
Revolution and Indicated Horse-power Curve
constructed from data of progressive runs
over measured mile course off
PROVINCETOWN, MASS
May - 29 - 1905
Plate 2
Revolutions
Horse Power
Curve of Indicated Horse Power
Curve of Revolutions
Engine Speed

Table V.—U. S. S. *GALVESTON.*

Data deduced from Progressive Runs over Measured-Mile Course off Provincetown, Mass., May 29, 1905.

Speed in knots per hour.	Revolutions per minute.	Indicated horsepower.			Slip of propeller in per cent. of its own speed.	Constant, R ÷ S.
		Starboard engine.	Port engine.	Total both engines.		
6	61.5	90	90	180	15.55	10.25
6.1	62	91	91	182	14.83	10.162
6.2	63	94	94	188	14.80	10.162
6.3	63.8	96	96	192	14.51	10.127
6.4	64.7	100	100	200	14.36	10.110
6.5	65.6	102	102	204	14.22	10.093
6.6	66.5	105	105	210	14.08	10.075
6.7	67.4	108	108	216	13.94	10.060
6.8	68.3	111	111	222	13.80	10.045
6.9	69.2	115	115	230	13.68	10.030
7	70.2	120	120	240	13.68	10.028
7.1	71.1	123	123	246	13.55	10.015
7.2	72.1	126	126	252	13.55	10.015
7.3	73.1	130	130	260	13.55	10.013
7.4	74.1	135	135	270	13.54	10.013
7.5	75.2	140	140	280	13.66	10.026
7.6	76.2	145	145	290	13.66	10.027
7.7	77.3	150	150	300	13.76	10.038
7.8	78.3	158	158	316	13.76	10.038
7.9	79.4	163	163	326	13.87	10.050
8	80.5	169	169	338	13.96	10.082
8.1	81.5	175	175	350	13.95	10.060
8.2	82.6	180	180	360	14.06	10.072
8.3	83.7	185	185	370	14.15	10.083
8.4	84.8	195	195	390	14.24	10.095
8.5	85.8	202	202	404	14.22	10.095
8.6	87	210	210	420	14.43	10.117
8.7	88.1	217	217	434	14.51	10.127
8.8	89.2	226	226	452	14.59	10.135
8.9	90.2	234	234	468	14.58	10.135
9	91.4	242	242	484	14.76	10.155
9.1	92.5	252	252	504	14.84	10.165
9.2	93.6	260	260	520	14.91	10.174
9.3	94.8	270	270	540	15.07	10.193
9.4	95.8	280	280	560	15.07	10.190
9.5	97	290	290	580	15.22	10.210
9.6	98.1	302	302	604	15.28	10.220
9.7	99.3	315	315	630	15.43	10.236
9.8	100.4	330	330	660	15.50	10.245
9.9	101.6	337	337	674	15.65	10.260
10	102.7	350	350	700	15.71	10.27
10.1	103.8	365	360	725	15.77	10.280
10.2	105	380	375	755	15.90	10.295
10.3	106.2	390	385	775	16.12	10.310
10.4	107.2	405	395	800	16.01	10.310
10.5	108.5	420	410	830	16.22	10.335
10.6	109.5	430	420	850	16.20	10.330
10.7	110.6	445	435	880	16.25	10.337
10.8	111.9	460	450	910	16.46	10.362
10.9	113	475	460	935	16.50	10.367

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Table V.—Continued.

Speed in knots per hour.	Revolutions per minute.	Indicated horsepower.			Slip of propeller in per cent. of its own speed.	Constant, R ÷ S.
		Starboard engine.	Port engine.	Total both engines.		
11	**114.3**	**495**	**475**	**970**	**16.69**	**10 390**
11.1	115.3	505	485	990	16.66	10.390
11.2	116 5	520	500	1,020	16.77	10.390
11.3	117.7	535	515	1,050	16.88	10.420
11.4	118.4	550	530	1,080	16 65	10.386
11.5	120	575	540	1,115	17 04	10.435
11.6	121.2	585	565	1,150	17.15	10.450
11.7	122 3	605	580	1,185	17.18	10.450
11.8	123.5	620	600	1,220	17.29	10.465
11.9	124.6	635	615	1,250	17.32	10 470
12	**125.9**	**655**	**630**	**1,285**	**17.48**	**10.490**
12.1	127.1	675	650	1,325	17.58	10.505
12.2	128.3	700	670	1,370	17.68	10.518
12.3	129.5	720	690	1,410	17.78	10.530
12.4	130 7	745	710	1,455	17.86	10.540
12.5	132	770	735	1,505	18.01	10 560
12.6	133.2	790	760	1,550	18.11	10.570
12 7	134.4	820	780	1,600	18.20	10.580
12 8	135 6	845	805	1,650	18.28	10.593
12.9	136.9	880	825	1,705	18.42	10.613
13	**138.2**	**905**	**850**	**1,755**	**18.58**	**10.630**
13.1	139.5	940	875	1,815	18.70	10.650
13.2	140.6	970	900	1,870	18.72	19.650
13 3	142	1,000	935	1 935	18.92	10.680
13.4	143 2	1,020	970	1,990	19.00	10.685
13.5	144 4	1,060	990	2,050	19 07	10.697
13.6	145.7	1,095	1,020	2,115	19.20	10.715
13.7	147	1,130	1,055	2,185	19 32	10.727
13 8	148.3	1,165	1,090	2,255	19.45	10.745
13 9	149.6	1,205	1,120	2,325	19 57	10.760
14	**150.8**	**1,240**	**1,150**	**2,390**	**19.64**	**10.770**
14 1	152.2	1,270	1,200	2,470	19.81	10.795
14.2	153 5	1,315	1,230	2,545	19 92	10.810
14.3	154.8	1,350	1,270	2,620	20 03	10.825
14.4	156.2	1,390	1,310	2,700	20.19	10.850
14.5	157.5	1,430	1,355	2,785	20.31	10.860
14 6	159	1,470	1,400	2,870	20.51	10.890
14.7	160.3	1,515	1,435	2,950	20.61	10.905
14.8	161.7	1,555	1,485	3,040	20.76	10.925
14.9	163.1	1,600	1,525	3,125	20 91	10.945
15	**164.5**	**1,645**	**1,570**	**3,215**	**21.06**	**10.967**
15.1	166	1,700	1,615	3,315	21.25	19.993
15.2	167.4	1,745	1,665	3,410	11.38	11.013
15.3	168.8	1,790	1,710	3,500	21.53	11.033
15.4	170.3	1,840	1,760	3,600	21.71	11.060
15.5	171.9	1,900	1,815	3,715	21 96	11.090
15.6	173.5	1,960	1,875	3 835	22.17	11.120
15.7	175	2,020	1,930	3,950	22.33	11.150
15.8	176 5	2,080	1,990	4,070	22.51	11.170
15.9	178.2	2,150	2,060	4,210	22.76	11.210
16	**180**	**2,225**	**2,130**	**4,355**	**23.06**	**11.250**
16.1	181.5	2,295	2,200	4,495	23.22	11.270
16.2	183.3	2,380	2,285	4,665	23.50	11.315
16.3	185	2,470	2,370	4,840	23.73	11.350
16.4	186.9	2,570	2,470	5,040	24 04	11.397
16.5	188.4	2,645	2,555	5,200	24.19	11 420
16.6	190.6	2,770	2,680	5,450	24 59	11.482

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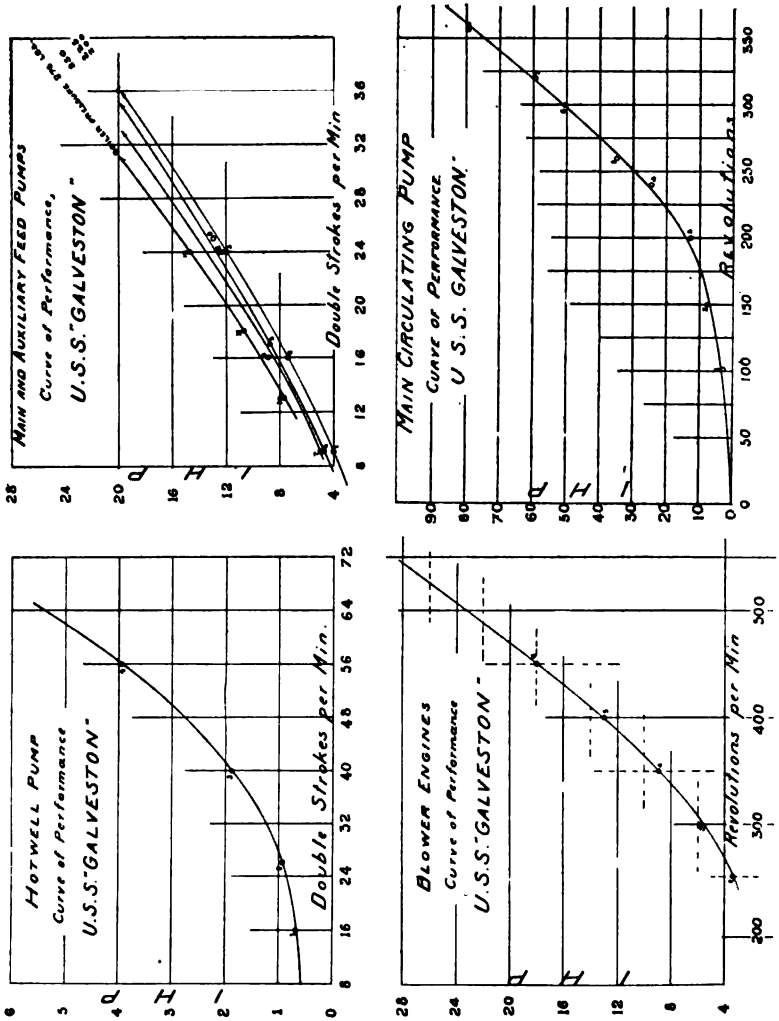

Plate III.—Performance Curves Auxiliary Machinery.

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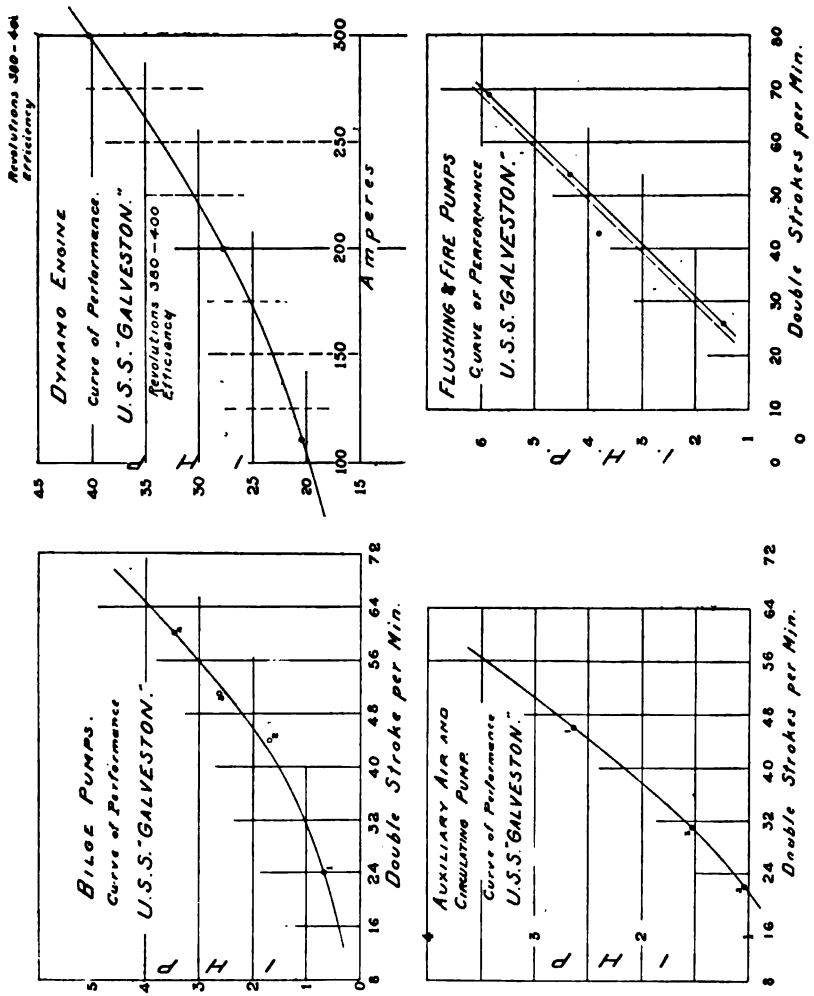

Plate IV.—Performance Curves Auxiliary Machinery.

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The draught at beginning was as follows:

Forward, feet and inches,	15–10½
Aft, feet and inches,	16– 4
Mean, feet and inches,	16– 1¼
Corresponding displacement, tons,	3,295

The time on the course was taken independently by four observers, and the revolutions checked by not less than three counters, of different types.

FOUR-HOURS' OFFICIAL TRIAL.

At 7·30 A. M. on May 30th the *Galveston* got under way from Provincetown and put to sea for the four-hours' full-speed trial required by the contract for the vessel.

The trial commenced at 10·30 A. M., the ship at the time being on a course N.N.W. ½ W. in a smooth sea; the weather conditions were good, calms and light airs prevailing throughout the trial.

At 12·30 P. M. a wide turn was made to the northward with two degrees of helm, and the vessel headed in toward Provincetown.

The draught and displacement at the beginning and end of trial were as follows:

	Beginning.	*End.*
Draught, forward, feet and inches,	15– 9	15– 6
aft, feet and inches,	16– 3½	16– 3
mean, feet and inches,	16– 0¼	15–10½
Displacement, tons,	3,280	3,230

The main and auxiliary machinery worked in a satisfactory manner; there was no undue vibration and no heating of bearings. The boilers steamed freely during the standardization run and the first three hours of the official trial.

The ship being in commission, the trial was run with the regular crew, a number of whom had only recently joined. The official trial was the first full-power run made.

The firing of the furnaces was controlled by an electric indicator; as the furnace numbers were shown, coal was thrown

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on the front of the grate and immediately afterwards the coals in an equal number of furnaces were leveled with a devil's claw or hoe. Slice bars were used occasionally through the slicing doors.

Steam pressure at the engine was limited to 250 pounds, but the safety valves on boilers were permitted to be set at 289 pounds. The air pressure in fire rooms was limited to two inches of water.

The mean average revolutions for the four hours corresponded to a speed of 16.406 knots per hour, while for the first three hours this amounted to 16.54 knots.

The following is a summary of the data recorded.

PERFORMANCE.—FOUR-HOURS' OFFICIAL TRIAL.

Steam Pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Steam pressure at boilers (per gauge), pounds	263	
engines	247	248
H.P. steam chest (gauge), pounds	240	238
1st receiver (absolute), pounds	96.6	102
2d receiver (absolute), pounds	31.7	31.5
Vacuum in condensers, inches of mercury	27.5	27

Temperatures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Injection, degrees	45.4	45.4
Discharge, degrees	85.7	83
Hotwell, degrees	89.6	72
Feed-water, degrees	186	183
Engine room, upper platform, degrees	85	91
working platform, degrees	85	80
Firerooms, working level, degrees	83.1	

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Main engines (maximum for 15 minutes)	191.46	192.9
(minimum for 15 minutes)	174	176.1
(mean for trial)	185.89	187.96
average for both	186.925	
Pumps, main, air	Attached.	
circulating	183	184
feed (forward and after)	34.4	31
hotwell	49.4	26
bilge	66	57
water service	...	59
auxiliary, feed	25.7	25
Blower engines, starboard, forward	491	

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	Starboard.	*Port.*
Blower engines, port, forward	493	
after	440	
Speed of ship, in knots per hour	16.406	
Slip of propeller, in per cent. of its own speed, based on mean pitch and average speed	23.53	24.45
Air pressure in firerooms, inches of water	2	

Mean effective pressures in cylinders, in pounds per square inch. (*Averages of cards taken at half-hourly periods*).

	Starboard.	*Port.*
Main engines, H.P. cylinder	110.13	105.69
I.P. cylinder	48.09	49.875
F.L.P. cylinder	14.73	14.825
A.L.P. cylinder	14.78	14.824
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	44.755	44.853

INDICATED HORSEPOWER.

Main engines, H.P. cylinder	775.71	752.7
I.P. cylinder	883.50	926.53
F.L.P. cylinder	407.38	414.58
A.L.P. cylinder	408.76	414.55
Total	2,475.35	2,508.36
both engines	**4983.71**	

AUXILIARIES.

Circulating pumps, main	11.1	11.2
Feed pumps (main and auxiliary)	67.42	
Hotwell pumps	2.85	1
Bilge pumps	4.13	3.13
Water-service pump	4.84	
Auxiliary condenser pumps	1.17	
Blower engines (forced draft)	62.25	
Dynamo engine (one in operation)	25.67	
Total	**194.76**	
all machinery in operation	**5,178.47**	

COAL.

Kind and quality	Pocahontas, run of mine.
Pounds per hour	10,022

DEDUCED DATA.

I.H.P. (total) per square foot of grate surface	17.26
heating surface	.3923
Pounds of coal per I.H.P. per hour (main engines, circulating, feed and hotwell pumps)	1.974
all machinery in operation,	1.935
Pounds of coal per square foot of grate surface	33.41
heating surface	.758
Cooling surface (main condenser), square feet per (total) I.H.P.	1.238
Heating surface (boilers), square feet per (total) I.H.P.	2.549

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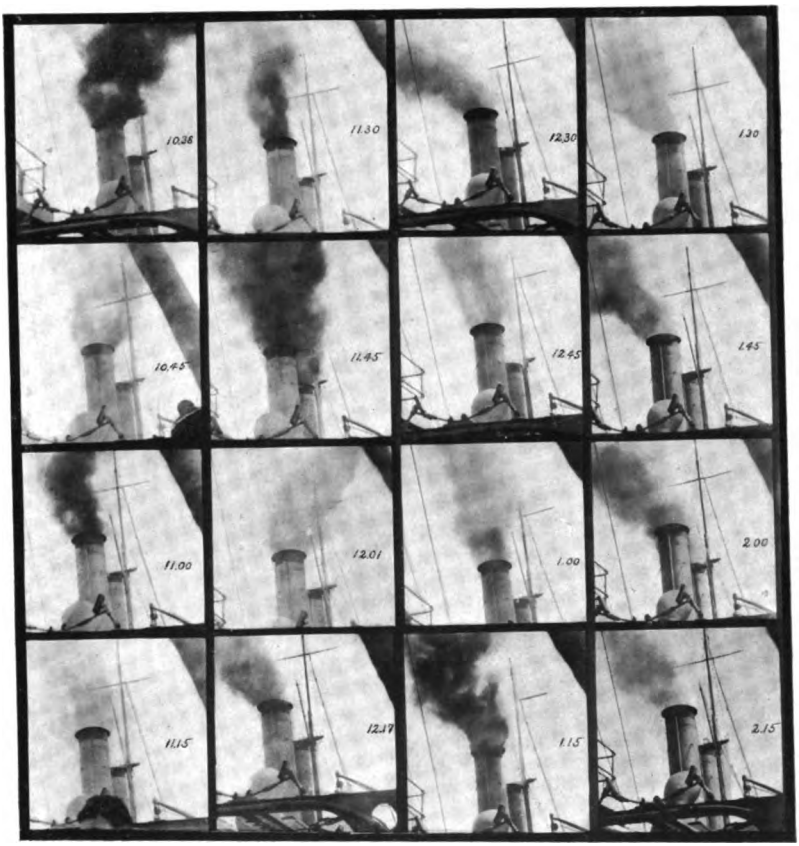

PHOTOGRAPHS OF SMOKE TAKEN DURING TRIAL, U. S. S. "GALVESTON."

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Table VI.—U. S. S. *GALVESTON.*

Performance on 24-hour Endurance Trial, May 31 to June 1, 1905.

.....	1st period, 11 A.M. to 11 P.M. May 31, 1905.		2d period, 11 P.M. May 31, to 11 A.M. June 1, 1905.		1st and 2d periods combined, 11 A.M. May 31 to 11 A.M. June 1, 1905.	
	Starboard.	*Port.*	*Starboard.*	*Port.*	*Starboard.*	*Port.*
Number of hours	12		12		24	
Draft	Natural.		Forced.		Natural and forced.	
Mean revolutions of engines	152.97	153.1	160.06	160	156.5	156.5
both engines	153.03		160.03		156.5	
Steam pressure in boilers above atmosphere, pounds	233.58		226.25		229.4	
H.P. cylinder above atmosphere, pounds	228.58	226.5	224.17	221.66	226.4	224.1
first receiver, absolute, pounds	61.78	59.6	69.87	65.95	65.8	62.8
second receiver, absolute, pounds	23.95	22.1	28.7	24.8	26.3	23.45
Steam cut-off in H.P. cylinder, inch	0.52		0.52		0.52	
I.P. cylinder, inch	0.56		0.56		0.56	
L.P. cylinders, inch	0.5		0.5		0.5	
Vacuum, inches	27.58	27.66	27.54	26.5	27.6	27.64
Temperature, fireroom, degrees Fahrenheit	125.25		97		108.3	
injection, degrees Fahrenheit	49.92	49.66	53.83	53.33	51.9	51.5
discharge, degrees Fahrenheit	89.66	80	91.66	83.66	90.7	81.8
feed water, degrees Fahrenheit	163.5	156	156.9	159.33	160.2	157.7
Revolutions of air pumps per minute	Attached.		Attached.		Attached.	
circulating pumps per minute	75	115	116.1	119.6	95.5	117.4
Grate surface in use, square feet	300		300		300	
I.H.P. main engines	1,234.9	1,280.29	1,385.4	1,414.9	1,347.59	1,310.15
both engines	2,515.19		2,800.3		2,657.74	
auxiliaries, estimated	160		160		160	
Aggregate I.H.P.	2,675.19		2,960.3		2,817.74	
Pounds of coal per hour	5,167.75		5,692.5		5,430.12	
per square foot of grate in use	17.226		18.975		18.1	
I.H.P., main engines	2.054		2.032		2.043	
and auxiliaries	1.931		1.923		1.927	
Mean draught at beginning of trial, feet and inches	15–10½		15–10½		15–10½	
Displacement, tons	3,230		3,230		3,230	
Mean speed by patent log in knots per hour	14.49		15.13		14.81	
speed curve in knots per hour	14.14		14.7		14.44	
Slip of screw, per cent	16.77		17.26		16.89	
Average air pressure, inches of water	None.		0.375		0.1875	

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A METHOD OF CALCULATING THE COAL ENDURANCE AND STEAMING RADIUS OF WAR VESSELS UNDER VARIOUS CONDITIONS.

By Lieut. Commander B. C. Bryan, U. S. Navy, Member.

The difficulty of procuring a supply of coal in time of war, and the necessity of steaming long distances without coaling, makes it important that a commanding officer should know the probable coal consumption and steaming radius of the vessel under his command.

The writer, a few years ago, spent considerable time collecting and tabulating the data at hand concerning U. S. naval vessels, with the object of forming a guide to enable the senior engineers and commanding officers of such vessels to tell what results might be expected under various conditions.—(See Journal of American Society of Naval Engineers, Vol. XIII, p. 50.)

Other articles upon this subject have appeared in this journal, both before and after the above compilation was published.—(See "Economic Marine Propulsion," John Lowe, Vol. III, p. 218; "Coal Endurance and Machinery of New Cruisers," I. N. Hollis, Vol. IV, p. 637; "Economic Speed and Power of War Vessels," &c., Vol. V., p. 66; "Speed and Coal-Consumption Curves," D. S. Mahony, Vol. XV, p. 366.)

The subject has been very ably treated in the above articles, and it is only with the idea of presenting in a practical way a method of tabulating and using the data collected that this article is written.

The Bureau of Steam Engineering has recognized the importance of collecting the necessary data from long runs of all vessels, and to this end has added a "Coal Endurance Form" in the back of the steam log, to be filled in for all continuous runs of seventy-two hours' duration or more. Although the results of a number of such runs are now on file, many of them are so incomplete or contain such glaring inaccuracies that it

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is often impossible to account for the erratic performances tabulated. In order to collect such data so that it may be of value, care and co-operation from those concerned is required; the coal and revolutions must be accurately recorded in the steaming log, together with all data concerning steam-using appliances, the speed through the water, weather conditions, trim and all conditions influencing the results should be accurately recorded in the deck log; thus co-operation between the navigating, engineer and deck officers is essentially necessary.

The problem presented to a commanding officer or senior engineer of a vessel is as follows: "The ship having come into port with the bottom more or less fouled, requiring a certain number of revolutions, exceeding the number required with a clean bottom, to make a certain speed, what would be the probable coal consumption at any other speed under the same conditions and what the most economical speed?" To answer this, in a practical way, is, as above stated, the object of this article.

From extensive investigation of the data on file for U. S. naval war vessels, it is believed that the coal consumption follows very closely the formula deduced by Chief Engineer John Lowe, U. S. Navy (now Rear Admiral, Retired), and given by him in his article in the JOURNAL of the Society, Vol. III, pp. 218–221, as follows: $Y=K+CR^3$, when

Y = total coal used for all purposes in tons per day;

R = average revolutions per minute of the main engines;

K = a constant, representing the coal in tons per day necessary for auxiliary purposes plus that necessary for keeping the main engine warmed when making zero revolutions;

C = a quantity, constant for given conditions.

Determination of the constant K should be made by a long trial or trials at the dock, the longer the better, with the auxiliaries in use as near as possible the average under usual conditions and the engines running as slowly as it is possible to run them. This will give the value of K under the usual conditions, and it may be corrected for other conditions by observations of the additional amount of coal necessary to produce those conditions by trials made while lying in port.

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The valve of C should be determined by the results of one or more long runs, under average conditions if possible. The results of all long runs of the vessel while in commission should be carefully tabulated on forms similar to Table I (or the coal consumption form to be found in back of steam log), and from these results C may be corrected from time to time, and other curves drawn to suit particular or peculiar conditions.

If no trial has been made to determine the value of K, an approximation may be obtained by selecting a number of actual runs and taking the value of R at several different points and the corresponding value of Y (the total coal per day), then assuming different values of C and subtracting the values of CR^3 from the corresponding values of Y until all values of the differences become the same or vary the least, the average difference then is an approximation for K. In doing this care should be taken that the data has been under normal conditions and accurately taken, and it is best to take an average of several runs for each value of Y and R. In case the value of K is found to be less than the average amount of coal used in port or less than the value of N_2 as found by the method given by Prof. I. N. Hollis in his article on "Economical Speed and Coal Endurance of War Vessels," &c., published in the JOURNAL, Vol. V, pp. 79 and 80, it may be taken for granted that the data is inaccurate.

For finding the relation of speed to revolutions a progressive trial should be made over a measured course and the usual data accurately recorded. The vessel should be clean when this trial is made, and the results will be the best that may be expected.

The value of the slip for each speed is now determined from these results.

To obtain the corresponding revolutions for any speed under different conditions of foulness, it is assumed that the slip will vary in the same proportion for each speed, that is, that if it is found that with a certain speed S when entering a port a certain number of revolutions R^1 is required giving a slip of Φ^1, the number of revolutions for the same speed with a

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clean bottom being R and the slip being Φ, then $\frac{\Phi^1}{\Phi}$ is a constant for all speeds, under the same conditions.

We then have $$\frac{P \times R - S \times \frac{6{,}080}{60}}{P \times R} \times 100 = \Phi, \text{ or}$$

$$P \times R\,(100 - \Phi) = S \times 10{,}133.3,$$

Similarly, $$P \times R^1\,(100 - \Phi^1) = S \times 10{,}133.3, \text{ or}$$

$$P \times R\,(100 - \Phi) = P \times R^1\,(100 - \Phi^1)$$

$$R^1 = R\left\{\frac{100 - \Phi}{100 - \Phi^1}\right\}.$$

From this the number of revolutions for any speed may be calculated for the existing conditions, the number of revolutions for one speed being known for the same conditions, and the equation of coal consumption $Y = K + CR^3$ will immediately allow the calculation of the probable coal consumption at the new speed.

The method recommended to be pursued is as follows:

Records of all long trips should be kept on blanks similar to Table I. The coal should be accurately recorded, and speed should be that which is made by the ship through the water; any error in the log should be allowed for and current should be eliminated. Notes should be made on each run of any occurrence or the use of any extraordinary amount of machinery, extra water distilled, or any use of steam that would affect the result. After a number of runs have been recorded, if in the meantime the trials for finding K have not been made, the curve is to be fitted to the average of the results as described. In doing this it is well to take the average of several runs at about the same speed for each point, and to take at least four points, one being as low as possible and another as high as it is likely long runs will be made.

Judgment must be used in selecting the runs. Short runs, and particularly those made for obtaining speed records for a few hours with limited auxiliaries in use, should be avoided, also runs in squadron, when reserve power is necessarily re-

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quired with corresponding extra coal consumption. From these results the curve of coal consumption may be laid down on revolutions, the easiest way to do this being probably to assume revolutions varying by 10 and find the value of Y from the equation $Y=K+CR^3$.

If a progressive trial has not been made, values of the speeds and revolutions for runs with a clean bottom should be selected; the length of the run is not of first importance if the speed and revolutions have been accurately recorded under normal conditions; the contractor's trial will usually give one point, and several others should be taken from which a curve of slip with a clean bottom may be laid down and the slip determined under these conditions for every speed, from which the corresponding revolutions may be calculated by the usual formula.

Curves of speed on revolutions may now be laid down for any increased value of the slip, assuming that the slip is increased in the same ratio for all speeds. Knowing the bunker capacity, curves of steaming radii for each value of the slip ratio may be added. As tables are considered more desirable by some people, such tables may be made out for each assumed value of the slip ratio.

To illustrate the method to be employed, the results as tabulated for the battleship *Kearsarge* and cruiser *Cleveland* have been laid down and tables made out for several values of the slip ratio, and are appended. To illustrate the use of the same, it is supposed that the *Kearsarge* came into port making 10 knots with 71 revolutions, and it is desired to know what the probable coal consumption would be under the same conditions going out at 12 knots. Following the 10-knot horizontal line on curve diagram we find it strikes 71 revolutions on the curve of 100 per cent. increased slip ratio; following this curve to 12 knots we find it will take 87.5 revolutions. Following down the ordinate to the coal consumption, we find it takes 125 tons per day, giving a steaming radius of 3,623 knots, or the same results may be found by consulting Table III, C.

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Table I.—U. S. S. *KEARSARGE.*

Kind of Coal, Various. Bunker Capacity, 1,573 Tons.

Number of run.	Number of hours' duration.	Speed in knots per hour.	Average revolutions per minute.	Total coal per day.	Steaming radius.	Date.	Days out of dock.	Number boilers in use.	Average steam pressure.	I.H.P.	Draught of vessel. F.	Draught of vessel. A.	REMARKS.—Cruising ground. Force and direction of wind and sea. Quality of coal. Number of fires banked. And any unusual amount of auxiliary machinery in use or any occurrence that will affect result.
				Tons.	*Knots.*	*1901.*			*Pounds*				
5	95	9.734	63.8	67.3	5,450	Dec. 28	24	3	...	1,927	...	...	Havana to Culebra.—Light breeze on bow. Sea smooth. New River coal.
						1902.							
6	96	7.842	51.9	53.8	5,509	March 5	90	4	...	1,194	...	...	Cienfuegos to Colon.—Light wind forward beam. Sea smooth. Pocahontas coal.
7	119	7.75	62.8	65.92	5,078	March 15	100	4	...	2,124	...	...	Colon to Trinidad.—Fresh breeze on bow. Moderate sea on bow. Pocahontas coal.
8	96	9.9	64.6	67.4	5,536	Nov. 17	19	4	...	2,245	...	...	Hampton Roads to Culebra.—Light breeze on bow. Medium sea abeam. Eureka coal; fair quality.
						1903.							
9	72	9.9	64.7	69.3	5,378	Feb. 12	105	5	...	2,210	...	...	Ponce to Galveston.—Light breeze on quarter. Moderate sea on quarter. George's Creek coal; fair.
10	72	10.9	74.2	87	4,719	Feb. 15	108	5	...	3,348	...	...	Ponce to Galveston.—Light breeze on quarter. Moderate sea on quarter. George's Creek coal; fair.
11	95	10.87	75.9	96.5	4,247	April 25	150	5	147	3,465	...	...	Pensacola to Cape Charles.—Wind variable. Smooth sea, to rough and variable. Poor Pocahontas coal.
12	165	12.1	78.5	98	4,656	June 6	16	5	145	3,555	...	...	Pensacola to Southampton.—Wind light on bow. Sea ahead moderate. Eureka coal; fair quality.
14	220	13.1	87.5	124.9	3,838	July 20	60	5	124	5,191	...	...	Southampton to Bar Harbor.—Wind and sea variable. Best English Admiralty Cardiff coal.
15	111	10.67	69	76.9	5,146	Dec. 4	15	4	156	2,689	...	...	Staten Island to Guantanamo, Cuba.—Eureka coal of good quality.
						1904.							
17	71	12	81	108.9	4,089	May 7	180	4–5	165	4,293	...	...	Pensacola to Guantanamo.—Good Pocahontas coal.
18	95	11	72	86.2	4,735	May 22	195	3	169	2,937	...	...	Guantanamo to Horta Fayal.—Good Pocahontas coal.
20	94	7.13	47.8	57	4,620	Aug. 6	270	3	160	1,188	...	...	Fiume, Austria, to Gibraltar.—Fair Imperial Navigation coal.
21	118	9.67	64.8	74.5	4,777	Aug. 15	279	4	160	2,550	...	...	Gibraltar to Horta Fayal.—Imperial Navigation and Ferndale coal.
22	194	10.28	68.9	73.8	5,168	Aug. 24	290	4	160	2,881	...	...	Horta Fayal to Newport, R. I.—Ferndale and Pocahontas coal.

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Table II.—U. S. S. *KEARSARGE.*

Data record of actual runs.						Comparison of actual and calculated data.	
No. of run.	Number of hours.	Revolutions. For run.	Revolutions. Average.	Coal per day. For run.	Coal per day. Average.	Coal per day from equation.	Difference from actual results (column 3).
14	220	87.5	84.3	124.9	116.9	125.1	+0.2
17	71	81		108.9		107.1	—1.8
12	165	78.5	75.1	98	91.9	100.9	+2.9
11	95	75.9		96.5		94.8	—1.7
10	72	74.2		87		91.1	+4.1
18	95	72		86.2		86.5	+0.3
15	111	69	65.5	76.9	70.7	80.7	+3.8
22	194	68.9		73.8		80.5	+6.8
21	118	64.8		74.5		73.4	—1.1
9	72	64.7		69 3		73.2	+3.9
8	96	64.6		67.4		73	+5.6
5	95	63.8		67.3		71.7	+4.7
7	119	62.8		65.9		70.2	+4.3
6	96	51.9	50	53.8	55.4	56.2	+2.4
20	94	47.8		57		52.2	—4.8

$R =$		50	65.5	75.1	84.3
$R^3 =$		125,000	263,372	423,565	599,078
$Y =$		55.4	70.7	91.9	116.9
$c =$	.00013	16.2	34.2	55.1	77.9
		39.2	36.5	36.8	39
					36.8
					36.5
					39.2
					4/151.5
					37 875

$Y = 38 + .00013\ R^3$.

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Table III.—U. S. S. *KEARSARGE*.

Bunker Capacity, = 1,573 tons.

Coal-consumption curve for natural draft, $V = 38 + .00013\ R^2$.

A.—PROGRESSIVE TRIAL RESULTS.

Speed in knots	4	5	6	7	8	9	10	11	12	13	14	15	16
Slip, per cent	...	...	...	7.64	7.85	8.22	8.94	9.25	9.8	10.5	11.1	11.88	...
Revolutions per minute	...	...	...	44.2	51	57.5	64	71.2	78	85.3	92.5	100	...
Coal per day, tons	...	...	...	49.2	55.2	62.7	72.1	84.9	99.7	118.7	140.9	168	...
Steaming radius, knots	...	...	...	5,371	5,471	5,418	5,235	4,892	4,543	4,135	3,751	3,370	...

Most economical speed = 52.7 revolutions = 8.27 knots. Steaming radius = 5,477 knots.

B.—SLIP 50 PER CENT. GREATER THAN PROGRESSIVE TRIAL.

Speed in knots	4	5	6	7	8	9	10	11	12	13	14	15	16
Slip, per cent	...	...	...	11.46	11.74	12.33	13.2	13.87	14.7	15.75	16.65	17.82	...
Revolutions per minute	...	...	...	46.1	53.3	60.2	67.3	75	82.5	90.6	98.7	107.2	...
Coal per day, tons	...	...	...	50.7	57.7	66.4	77.6	92.8	111	134.7	163	...	...
Steaming radius, knots	...	...	...	5,213	5,234	5,116	4,865	4,475	4,081	3,643	3,243	...	...

Most economical speed = 7.91 knots. Steaming radius = 5,238 knots.

C.—SLIP 100 PER CENT. GREATER THAN PROGRESSIVE TRIAL.

Speed in knots	4	5	6	7	8	9	10	11	12	13	14	15	16
Slip, per cent	...	...	...	15.28	15.7	16.44	17.88	18.5	19.6	21	22.2	23.8	...
Revolutions per minute	...	...	...	48.2	55.8	63.2	71	79.3	87.5	96.6	105.8	115.6	...
Coal per day, tons	...	...	...	52.6	60.6	70.8	84.5	102.8	125.1	155.2	...	...	...
Steaming radius, knots	...	...	...	5,023	4,984	4,800	4,468	4,040	3,623	3,162	...	...	...

Most economical speed = 7.6 knots. Steaming radius = 5,033 knots.

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Table IV.—U. S. S. *CLEVELAND.*

Kind of Coal, Various. Bunker Capacity, 668 Tons.

Number of run.	Number of hours duration.	Speed in knots per hour.	Average revolutions per minute	Total coal per day.	Steaming radius.	Date.	Days out of dock.	Number boilers in use.	Average steam pressure.	I.H.P.	Draught of vessel.		REMARKS.—Cruising ground. Force and direction of wind and sea. Quality of coal. Number of fires banked. And any unusual amount of auxiliary machinery in use or any occurrence that will affect result.
				Tons.	*Knots.*	*1904.*			*Pounds.*		*F.*	*A.*	
1	95	11.7	121.9	32.32	5,825	May 7	180	4	160	1,200	...	...	Pensacola to Guantanamo.—Light breeze on bow. Sea smooth. Pocahontas coal.
2	59	9.93	103.75	25.82	6,165	May 15	188	3	190	650	...	...	Guantanamo to St. Thomas.—Light air on bow. Sea smooth. Pocahontas coal.
3	117	10.32	107.08	28.27	5,861	May 20	193	4	205	...	...	...	St. Thomas to Horta Fayal.—Light breeze on bow. Sea smooth to medium. Cory's Merthyr coal.
4	107	10.15	102.7	24.14	6,740	May 26	199	4	...	...	...	...	St. Thomas to Horta Fayal.—Light breeze on bow. Medium sea. Cory's Merthyr coal.
5	71	13.49	134.8	38.32	5,644	June 1	204	4	200	...	...	...	Horta Fayal to Tangiers.—Light breeze aft. Sea smooth to medium. Cory's Merthyr coal.
7	73	10.1	105	25.2	6,426	Aug. 2	270	4	200	820	...	...	Corfu, Greece, to Ville France.—Light breeze on bow. Sea smooth. Excellent Cardiff coal.
8	111	10.6	105	25.2	6,680	Aug. 10	278	4	200	820	...	...	Ville France to Smyrna.—Light breeze on quarter. Sea Smooth. Excellent Cardiff coal.
9	149	10.56	106	26.4	6,400	Aug. 18	286	3–2	200	840	...	...	Smyrna to Gibraltar.—Light breeze on quarter. Sea smooth. Excellent Cardiff, mixed with poor Ferndale coal.
10	72	11.9	126.3	33.09	5,878	Sept. 5	304	4	200	1,250	...	...	Gibraltar to Cherbourg, France.—Light, variable breeze. Moderate sea. Excellent Pocahontas coal.
11	73	11.58	131.1	36.63	5,076	Nov. 3	363	4	200	2,000	...	...	Southampton to Gibraltar.—Light wind abeam. Sea smooth. Excellent Pocahontas coal.
12	195	12.76	131	39.2	5,210	Dec. 5	395	4	200	2,000	...	...	Gibraltar to Bridgetown, Barbadoes.—Light breeze on quarter. Sea smooth. Pocahontas coal.

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Table V.—U. S. S. *CLEVELAND.*

Data recorded on actual runs.						Comparison of actual and calculated data.	
No. of run.	No. of hours.	Revolutions. Average for run.	Revolutions. Average.	Coal per day. For run.	Coal per day. Average.	Coal per day from equation.	Difference from actual result.
5	71	134.8	132.3	38.32	38.05	39	+ 0.68
11	73	131.1		36.63		37	+ 0.36
12	195	131.1		39.2		37	— 2.2
10	72	126.3	124.1	33.09	32.7	34.6	+ 1.5
1	95	121.9		32.32		32.6	+ 0.28
3	117	107.08	104.9	28.27	25.84	27.7	— 0.43
9	149	106		26.4		26.4	± 0.
8	111	105		25.2		26.1	+ 0.8
7	73	105		25.2		26.1	+ 0.8
2	59	103.75		25.82		24.7	— 0.12
4	107	102.7		24.14		25.3	+ 1.16

R =	104.9	124.1	132.3
R^3 =	1,154,320	1,911,240	2,315,685
Y =	25.84	32.70	38.05
c = .00001			
cR^3 =	11.54	19.11	23.16
	14.30	13.59	14.89

Average for $K = 14.26$ Call $K = 14.5$

Coal consumption $Y = 14.5 + .00001 R^3$

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Table VI.—STEAMING RADIUS U. S. S. *CLEVELAND.*

Bunker Capacity, 668 tons. $V = 14.5 + .00001\ R^3$.

A.—PROGRESSIVE TRIAL, CLEAN BOTTOM.

Speed, knots	4	5	6	7	8	9	10	11	12	13	14	15	16	17
Slip, per cent	...	...	15.35	14.85	14.6	14.55	14.8	15.4	16.2	17.2	18.4	19.9	22	...
Revolutions per minute	...	...	61.28	71.08	81	91.05	101.5	112.4	123.8	135.75	148.3	161.9	177.4	...
Total coal per day, tons	...	...	16.8	18.1	19.8	22	25	28.7	33.4	39.3	48.1	59.9	...	...
Steaming radius, knots	...	...	5,725	6,200	6,478	6,558	6,413	6,145	5,760	5,302	4,666	4,015	...	...

Most economical speed = 89.8 revolutions = 8.88 knots. Steaming radius = 6,560.

B.—SLIP 50 PER CENT. MORE THAN CLEAN BOTTOM.

Speed, knots	...	...	6	7	8	9	10	11	12	13	14	15	...	...
Slip, per cent	...	...	23.02	22.31	21.9	21.83	22.2	23.1	24.3	25.8	27.6	...	...	...
Revolutions per minute	...	...	67.4	77.9	88.6	99.5	111.1	123.7	137	151.5	167.2	...	...	...
Total coal per day, tons	...	...	17.6	19.2	21.5	24.4	28.2	33.4	40.2	49.2	...	...	...	...
Steaming radius, knots	...	...	5,466	5,845	5,966	5,914	5,686	5,280	4,785	4,236	...	...	...	...

Most economical speed = 8.1 knots. Steaming radius = 5,984.

C.—SLIP 100 PER CENT. MORE THAN CLEAN BOTTOM.

Speed, knots	...	...	6	7	8	9	10	11	12	13	14	15	16	17
Slip, per cent	...	...	30.7	29.7	29.2	29.1	29.6	30.8	32.4	34.4	36.4	...	...	...
Revolutions per minute	...	...	74.9	86.1	97.7	109.7	122.8	137.4	160.1	...	...	...	...	...
Total coal per day, tons	...	...	18.7	20.9	23.8	27.7	33	40.4	55.5	...	...	...	...	...
Steaming radius, knots	...	...	5,144	5,369	5,389	5,208	4,858	4,365	3,466	...	...	...	...	...

Most economical speed = 7.3 knots. Steaming radius = 5,390.

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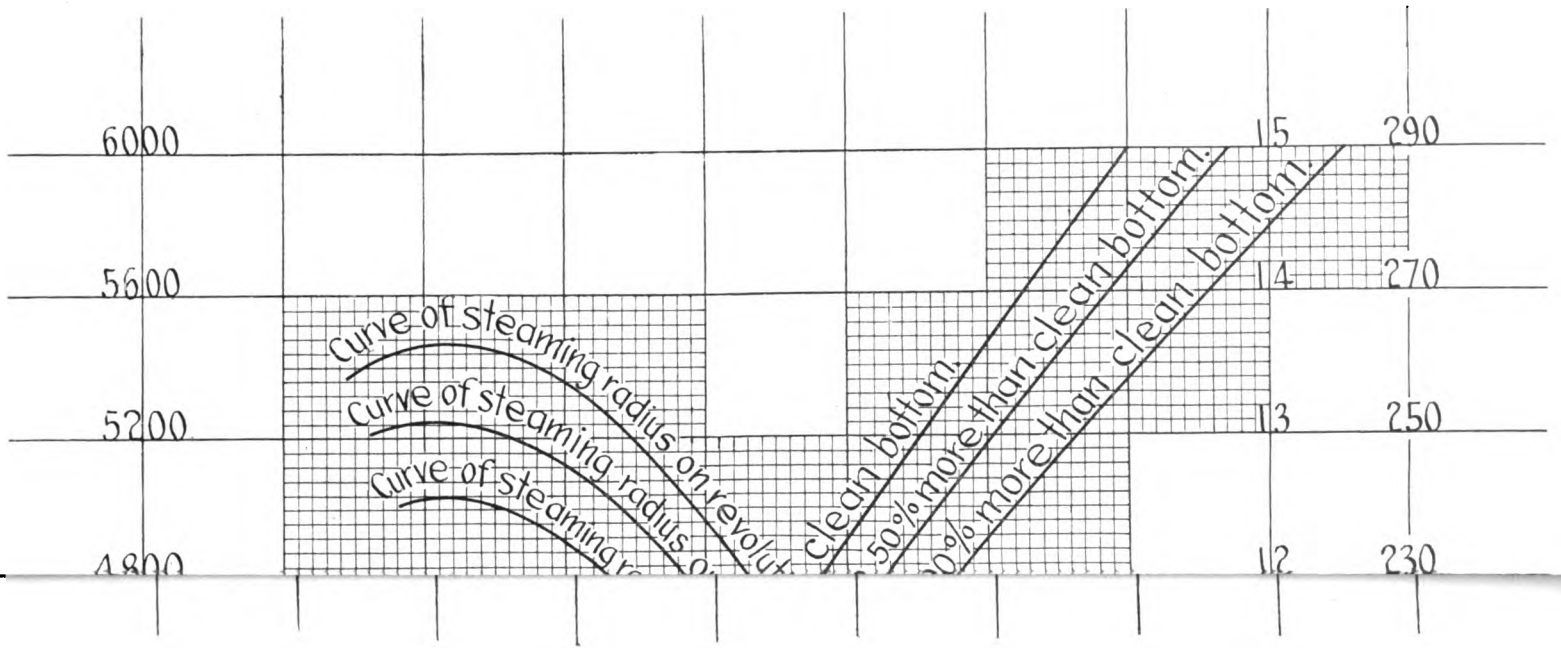

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A COMPARISON OF THE PERFORMANCES OF PROPELLERS U. S. S. *WYOMING.**

By Lieutenant J. P. Morton, U. S. Navy.

The steam engineering specifications for the U. S. S. *Wyoming* called for the manufacture of a spare set of propeller blades to be built after the vessel's trial and to conform to specifications that would be furnished by the Bureau.

These blades were designed with increased projected and expanded areas, increased diameter of propeller and decreased pitch, but after being in service the pitch was increased.

A comparison of the performances of the vessel with the original propellers and with the ones now in place, under the two conditions of pitch, may be of some interest as illustrating the great range of efficiency for a small change of pitch. The material point is not the improvement brought about by the change of blades, as that was generally anticipated, but in the startling effects produced by the change of pitch in the new blades. The new propellers were first installed with a pitch of 7 feet 6 inches. After about a year the pitch was changed to 8 feet 6½ inches. The following results will prove that this change was justifiable.

The original propellers were rights-and-lefts, three bladed, built-up, true screws, bent back five inches, and of manganese-bronze. The top tips of the blades turned inboard in the ahead motion. The pitch was adjustable from 7 feet 8.6 inches to 9 feet 4.4 inches.

The propellers now in use differed from the original in the particulars noted in the following table. Their pitch is adjustable from 7 feet 4 inches to 8 feet 6½ inches. The general appearance of the blades is shown in the sketches.

For the sake of brevity, the conditions enumerated in the following table will hereinafter be referred to as (1), (2) and

* For a description of the *Wyoming* see Vol. XV, page 83.

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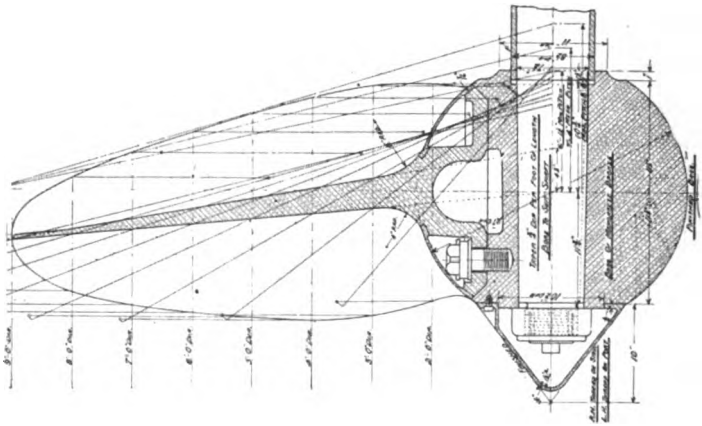

Fig. 2.—New Propeller.

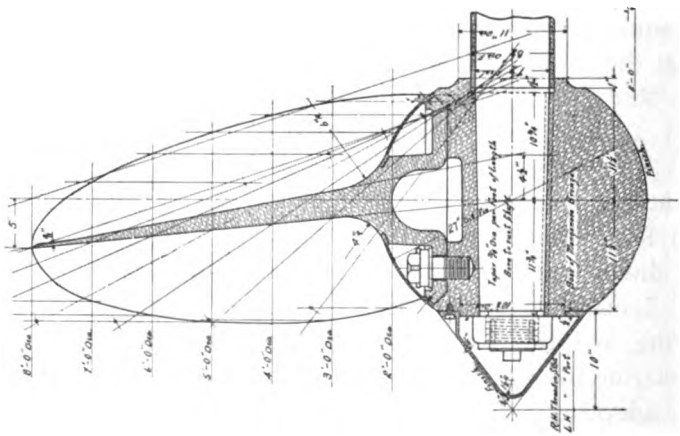

Fig. 1.—Original Propeller.

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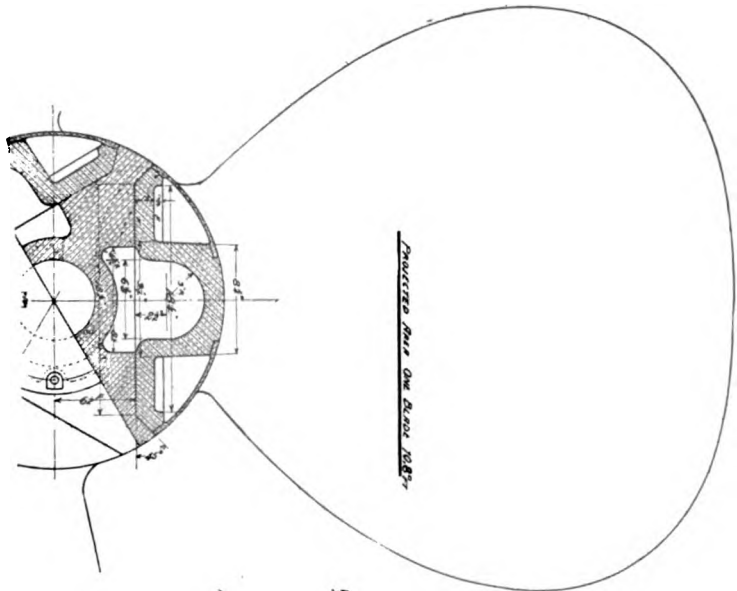

Fig. 2a.—NEW PROPELLER.

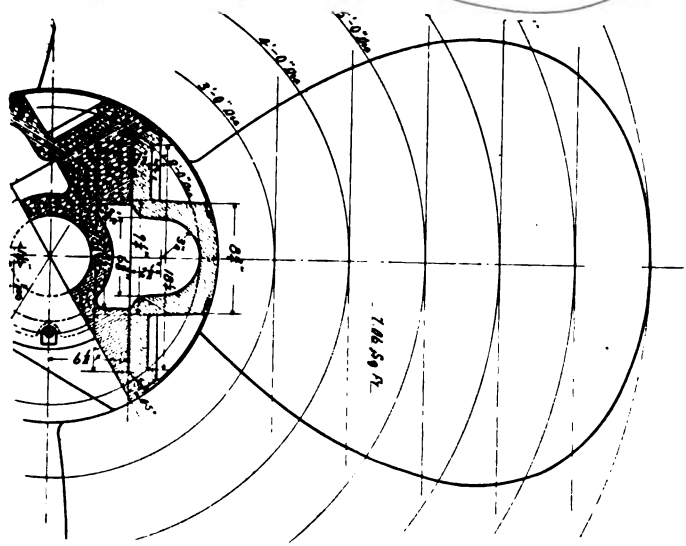

Fig. 1a.—ORIGINAL PROPELLER.

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Condition.	(1)	(2)	(3)
CA, ...	13.189	12.826	14.003
Advance of screw (ship), . .	6.0141	5.632	6.401
Reverse current, ...	7.1749	7.194	7.602

The above table is calculated for the highest speeds obtained under the various conditions, and in this, as in all following comparisons, it must be borne in mind that with the new propeller, condition (3), a higher speed was obtained than under the other two conditions. By resolving CA into its components and comparing them with CN we get the following table, giving the percentages of power employed in driving the ship ahead, driving the water astern, and in wastage athwartships:

Condition.	(1)	(2)	(3)
To advance of screw, . . .	35	38	37
Reverse current, ...	42	48	45
Wastage athwartships, . .	23	14	18

The increase of efficiency is especially noticeable in comparing (3) with (1), but the relatively low speed obtainable under (2) probably accounts for the rather unfavorable light in which (3) is viewed when compared with (2). This would seem to be dissipated by a study of the following table, which is worked out for various speeds, especially the common speed of 8 knots.

Revolutions.	H.P.	Speed. (1)	Advance of screw.	Reversed current.
91.5	211.55	5.97	6.612	6.557
124	495	8	6.344	6.825
134.2	589.99	8.57	6.4715	6.7175
170.5	1,295.5	10.4	6.1813	7.0077
210.3	.....	11.08	5.9403	7.2487
214.7	2,811.01	12.37	6.0141	7.1741

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Revolutions.	H.P.	Speed.	Advance of screw.	Reversed current.
		(2)		
104.9	265	6.5	5.844	6.982
129.5	475	7.6	5.947	6.879
134	557	8	6.04	6.785
153	810	8.7	5.826	7
176.3	2,120	10.3	5.927	6.899
208.7	2,770	11.6	5.632	7.194
		(3)		
75	220	5.29	8.26	5.743
100	300	6.93	7.022	6.981
117.5	460	8	6.899	7.104
125	600	8.61	6.977	7.026
149.7	1,000	9.9	6.688	7.215
174.5	1,730	11.4	6.62	7.383
198.5	2,610	12.54	6.401	7.602

NOTE.—The above results were obtained in trials over the measured mile. During trials (1) and (2) efforts were made to steady ship on a course with one engine stopped, but it was found to be impracticable. During (3) trial the following results were obtained:

	Revs.	Speed.	H.P.
Running starboard engine only	173.5	8.72	1,107.16
Running port engine only	184	8.18	1,518.06

The following data was obtained under condition (3) on August 28 and 29, 1904, during a sea run, under most favorable conditions. The coal was carefully weighed. The patent log was checked by later runs over the measured mile. The steaming radius was calculated from the coal consumption of the short runs which were of about four hours duration at each speed. The ship's bottom had recently been cleaned; the sea was smooth, and wind very light and from ahead.

Revs.	Knots.	Coal per day.	Steaming radius.	Slip.	H.P.
		Tons.	*Knots.*		
100	7.07	18.75	3,298	16.1	334.34
110	7.65	20.35	3,107	17.4	418.21
120	8.45	22.35	3,380	16.4	470.8
130	9.3	24.6	3,303	14.8	654.65
140	9.9	27.2	3,183	15.8	831.91
150	10.7	29.7	3,158	15.4	959.99
160	11.1	32.5	3,198	17.6	1,138.9

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TESTS OF THE CAVITE STEEL FLOATING DRY DOCK.

By Civil Engineer A. C. Cunningham, U. S. Navy, Member.

In the early part of June, 1905, the Maryland Steel Co. reported the Cavite self-docking steel floating dry dock ready for test, having completed the same in one month's less time than the twenty-seven allowed by the contract.

As in the case of the Government floating dock at New Orleans, built by the same company, the Cavite dock was constructed in a shallow basin, and on completion the basin was flooded and the cofferdam at the entrance removed. While flooding the basin the valves of the dock were left open and the dock allowed to fill, so that it continued to rest on the pile and timber foundation on which it was built until the time selected for floating and towing it to the mouth of the Patuxent River, the place selected for test.

When the dock rose from its timber bed it was christened the *Dewey*, after the Admiral of the Navy, by Miss Grace Endicott, the daughter of Rear Admiral M. T. Endicott, Chief of the Bureau of Yards and Docks, Navy Department, under whose administration the dock had been built, and who represented the Navy Department when the dock was floated.

In the preliminary test of the New Orleans floating dock a collier of 6,000 tons displacement was used. The increased confidence in this class of dock for warships, and in the *Dewey* dock in particular, is shown by the fact that one of the latest and heaviest ships, the armored cruiser *Colorado*, was selected for the preliminary test.

Before docking the *Colorado*, the dock was sunk to twenty-nine feet draught over the keel blocks and then pumped up light to a freeboard of two and one-half feet. The sinking was

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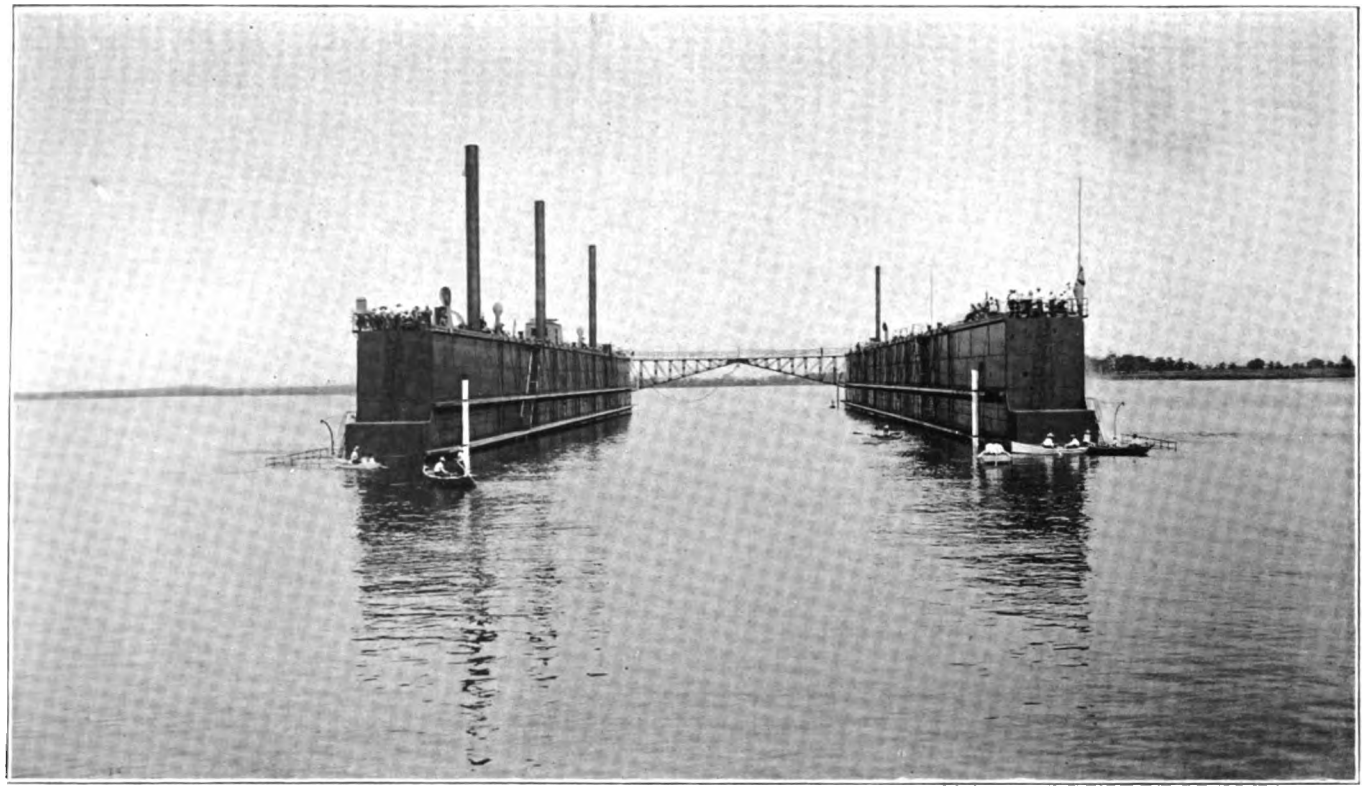

U. S. Floating Dry Dock "Dewey."—Dock Sunk Ready to Receive Ship.

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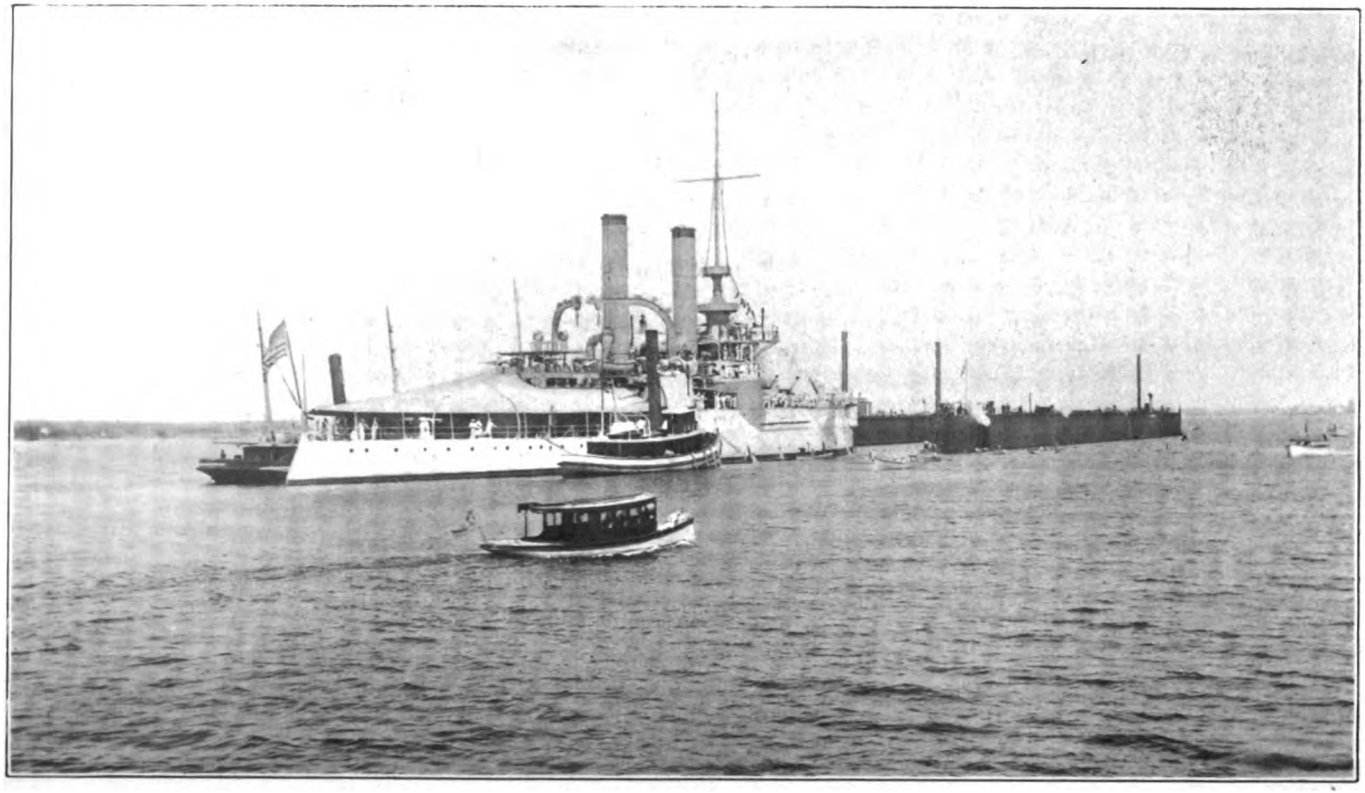

U. S. Floating Dry Dock "Dewey."—U. S. Battleship "Iowa" Entering Dock.

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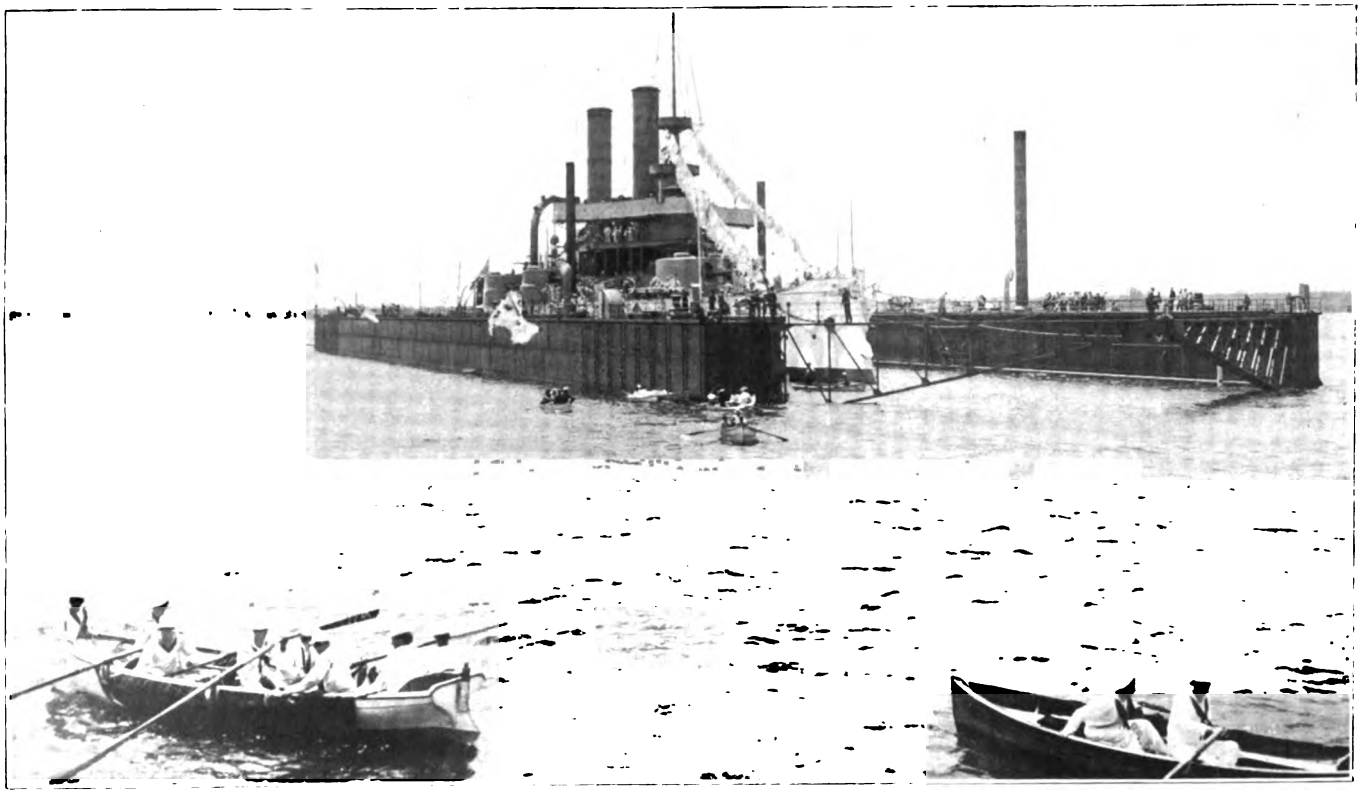

U. S. Floating Dry Dock "Dewey."—U. S. Battleship "Iowa" Centered and Ready for Lifting.

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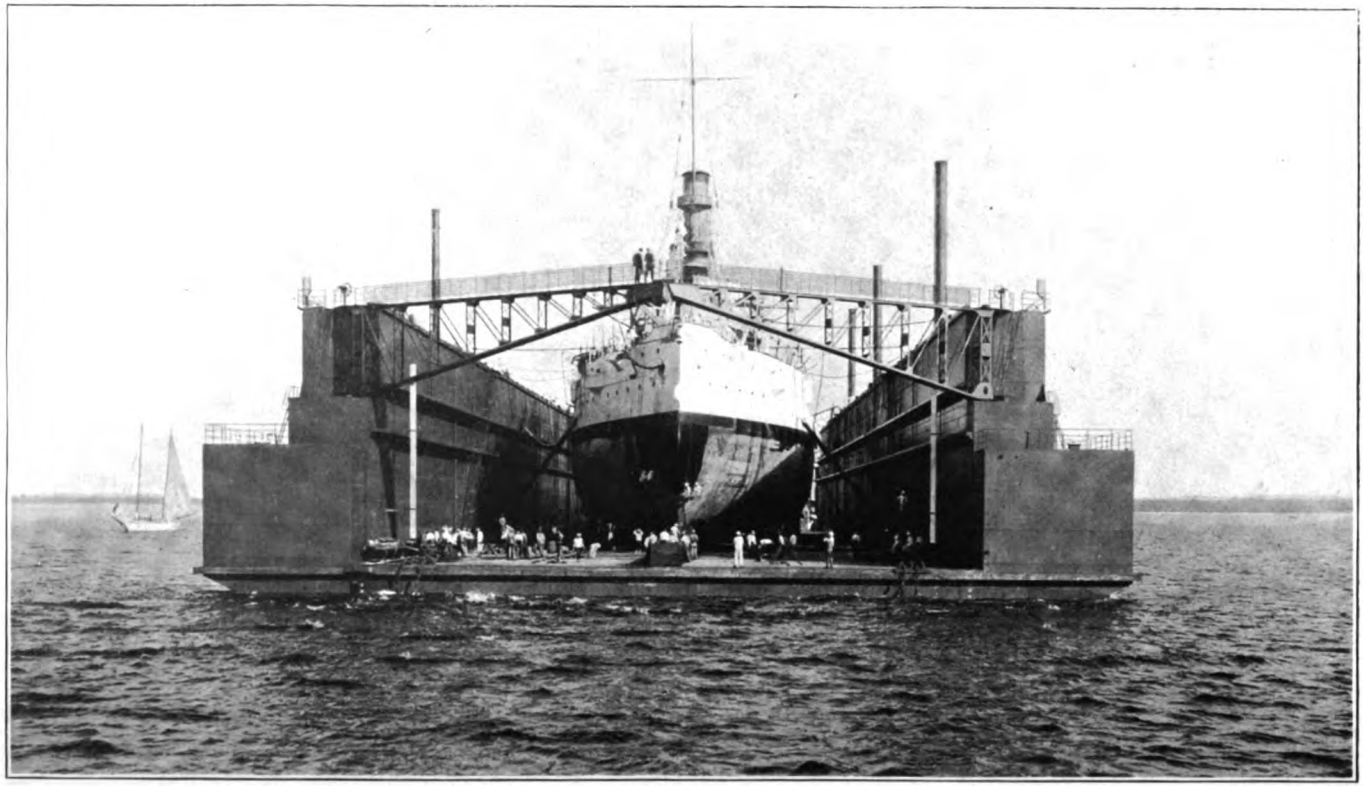

U. S. Floating Dry Dock "Dewey."—U. S. Battleship "Iowa" Completely Docked.

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accomplished in one hour and thirty-six minutes and the pumping up in one hour and two minutes, showing that the dock is of very rapid action. On this test it was also demonstrated that little or no trimming was required, the dock sinking and rising practically level with all valves open.

The U. S. S. *Colorado* was docked on June 23, 1905, having a displacement of 13,300 tons at that time. The main and docking keel blocks were all set at the same height. In this preliminary test no effort was made to secure speed, and one-half hour was used in making flushing and fire connections. The elapsed time from when the ship landed on the blocks until the keel came out of water was two hours and sixteen minutes. Pumping was continued until the dock had a uniform freeboard of two and one-half feet, only enough excess of water being retained in the side walls and end compartments to give the necessary trim. The *Colorado* was carried on the dock about twenty-four hours without changing the water ballast. When the dock had reached a freeboard of two and one-half feet with the *Colorado*, the deflection on the main keel line in the five hundred feet of length of the dock was about one-quarter of an inch; after about twenty-four hours the deflection in five hundred feet increased to about one and one-sixteenth inches. After undocking the *Colorado* the dock was found to have practically straightened without retaining any set.

After the undocking of the *Colorado*, deflection observations were continued for three days, and variations in deflections with the dock unloaded, due to temperature changes, of seven-eighths of an inch were noted.

The battleship *Iowa* was docked on June 27, 1905, for a record test, having a displacement of 11,600 tons at the time, and was carried on the dock for forty-eight hours. The specifications required that a 16,000-ton ship should be raised in four hours from the time the ship took the blocks until the keel was out of water. For the equivalent of a 16,000-ton ship the dock was pumped to a freeboard of four and one-half feet. From the time the *Iowa* took the blocks until the keel

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was out of water was one hour and thirty-seven minutes; to the time the dock had a freeboard of four and one-half feet, two hours and forty-two minutes. During the docking of the *Iowa* one of the three pumping engines was out of commission for forty-two minutes with a slipped eccentric, so that the actual time of operation of the dock is about half that allowed by the specification.

The *Iowa* was docked by uniform pumping, as in the case of the *Colorado*, and carried for forty-eight hours without change of water ballast in the dock. The specification required that when a ship had been docked by uniform pumping until the dock had a freeboard of two feet the deflection in the entire 500 feet of length of the dock should not exceed three inches. When the dock reached a freeboard of four and a-half feet with the *Iowa*, the deflection was about two inches. During the first twenty-four hours, the dock remaining uniformly pumped, the deflection increased to four inches in the 500 feet, and during the second twenty-four hours showed a recovery to three and three-eighths inches.

Immediately following the undocking of the *Iowa* the dock was pumped up to the same depth of water in the compartments as when the ship was docked, which gave a freeboard of nine feet and six inches, and it was found that the dock had a hog of one inch. During the night this hog disappeared, and early the next morning was a half inch sag. The greatest deflection in the bearing length of the *Iowa* while carried on the dock was about one and three-quarter inches. The deflection observations indicate that there was no permanent set caused by the docking, and that temperature variations may cause considerable hog or sag.

After the undocking of the *Colorado* the main and docking keel blocks were found to be uniformly indented about one-sixteenth of an inch with no crushing. No change was made in the blocks for the *Iowa*, and after undocking she was found to have rested even more easily than the *Colorado*.

With two feet of freeboard and one foot of water remaining in the pontoons the carrying capacity of the dock is 18,500

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tons, and a 20,000-ton ship could readily be docked with sufficient freeboard to admit of easily working on its bottom.

The results of the docking tests with the *Dewey* dock show that it is considerably in excess of the contract requirements in strength, time of operation and capacity, and is in all respects a very noticeable advance on all floating dry docks which have been so far projected or built. The greatest innovation is the requirement of uniform pumping. This not only insures safety from careless or unskilled handling, but makes it possible to dock nearly all ships with keel straight, or with as much hog or sag as circumstances may render desirable, by suitably distributing the water ballast in the dock. The ship may be also hogged or sagged while on the dock, should occasion arise for so doing. With thirty feet of water over four-foot keel blocks the side walls of the dock have a freeboard of eleven feet. By taking the keel blocks down to two feet and sinking until the side walls have a freeboard of three feet, forty feet of water may be had over the blocks, so that a ship may be taken into the dock in any condition of disablement.

Although a heavier ship, the deflections with the *Colorado* were less than with the *Iowa*, on account of the much longer bearing length of the *Colorado's* keel. With the uniform pumping the strains caused by the *Iowa* were much greater than will be caused by any other ship in the service, projected or built; so that the dock has had the most severe test that can be imposed on it.

The dock was constructed under the supervision of Civil Engineer Leonard M. Cox, U. S. N., and tested by a Board of naval officers, consisting of Captain Adolph Marix, senior member; Naval Constructor J. H. Linnard, Commander J. F. Parker, Commander W. F. Worthington, Naval Constructor J. H. Rock, Civil Engineer A. C. Cunningham and Assistant Civil Engineer J. S. Shultz. The Board was assisted in the tests by Civil Engineer Cox.

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THE SELF-DOCKING TESTS OF THE STEEL FLOATING DRY DOCK FOR CAVITE, P. I.

By Civil Engineer Leonard M. Cox, U. S. Navy.

In the specifications forming a part of the contract between the Government and the Maryland Steel Company for the construction of the steel floating dry dock for Cavite, P. I., it is provided that the structure, after completion, shall be given a thorough working test before final acceptance. This test, as provided for in the contract and its specifications, comprises three distinct operations: The lifting, within a given time, to a specified freeboard and with limited deflections, of the heaviest type of cruiser; the lifting, under similarly specified conditions, of the heaviest battleship; and, finally, the lifting of its own hull clear of water.

On June 10, 1905, the dock was launched from the building basin at the manufacturers' works and was immediately towed to the mouth of the Patuxent River, the site selected for the official tests. After preliminary trials of the machinery, the dock was declared ready for testing on June 17, 1905, just one month before the expiration of the contract time of twenty-seven months from the date of award. On June 23d, the cruiser *Colorado* was docked, followed by the battleship *Iowa* on June 27th, and preparations were at once begun for the self-docking trials.

An article appearing in this issue of the Journal (page 714) fully describes the successful docking of the *Colorado* and the *Iowa*. This article, coming as it does from the officer who wrote the specifications under which the dock was constructed and who was closely connected with its preliminary design, is of peculiar interest, but, as it covers only the ship-docking tests, it may not be out of place to add a brief account of the subsequent self-docking operations.

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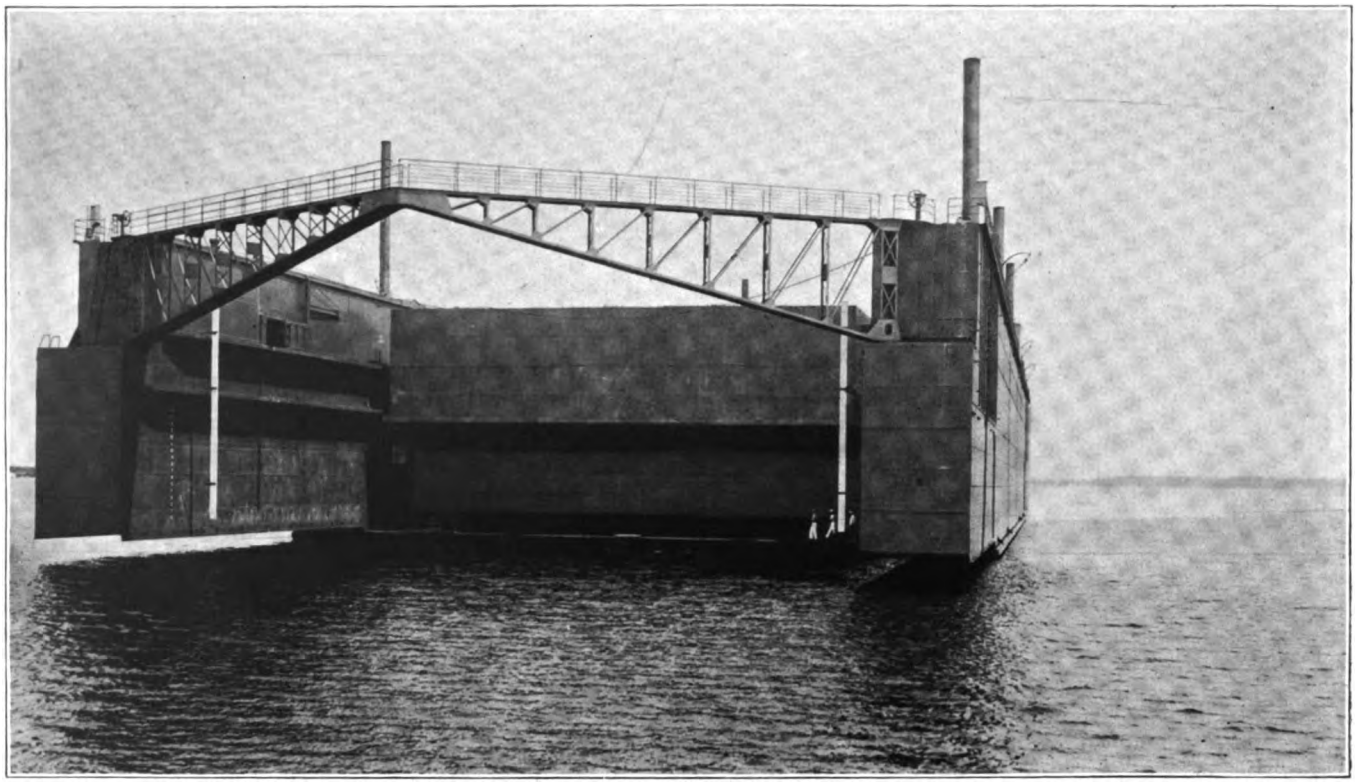

U. S. Floating Dry Dock "Dewey."—End Pontoons Self-Docked on Center Pontoon.

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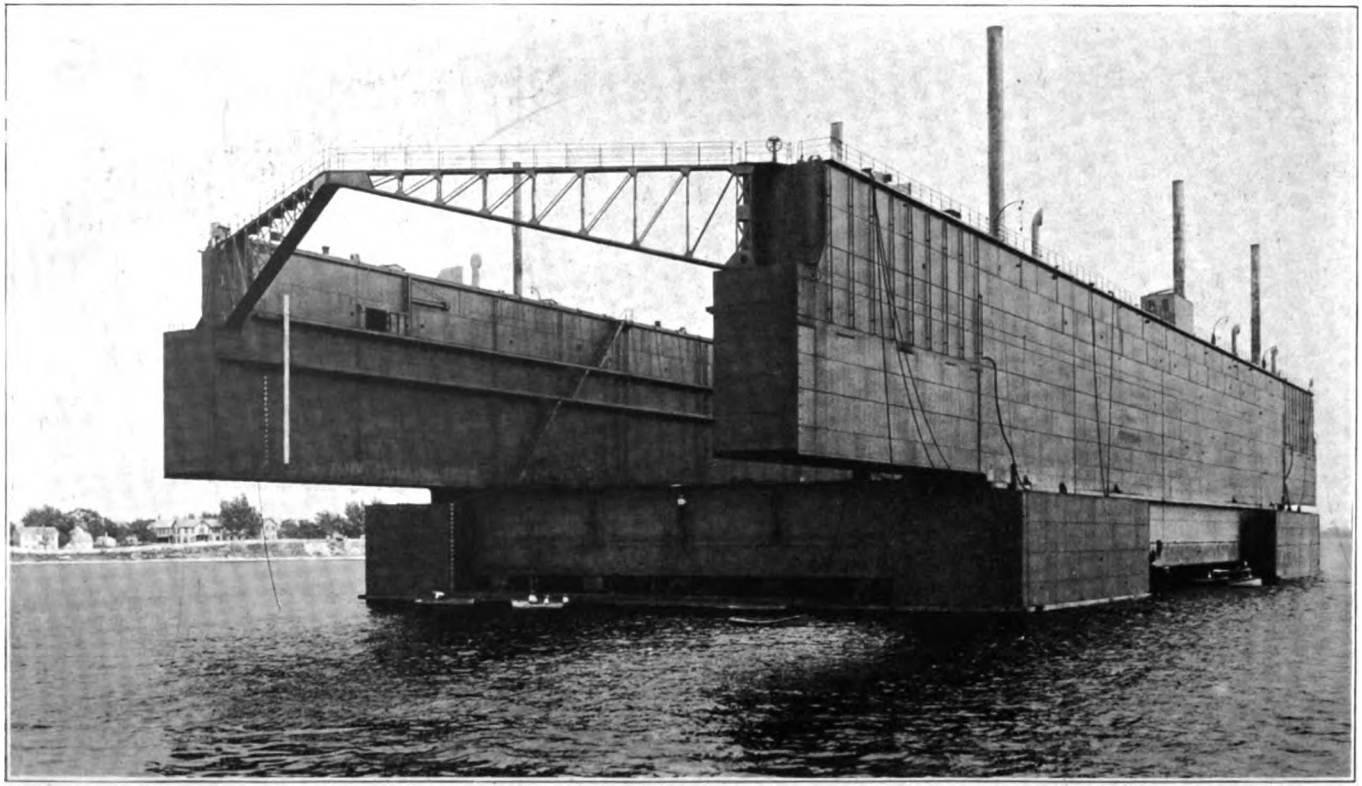

U. S. Floating Dry Dock "Dewey."—Center Pontoon Self-Docked on End Pontoons.

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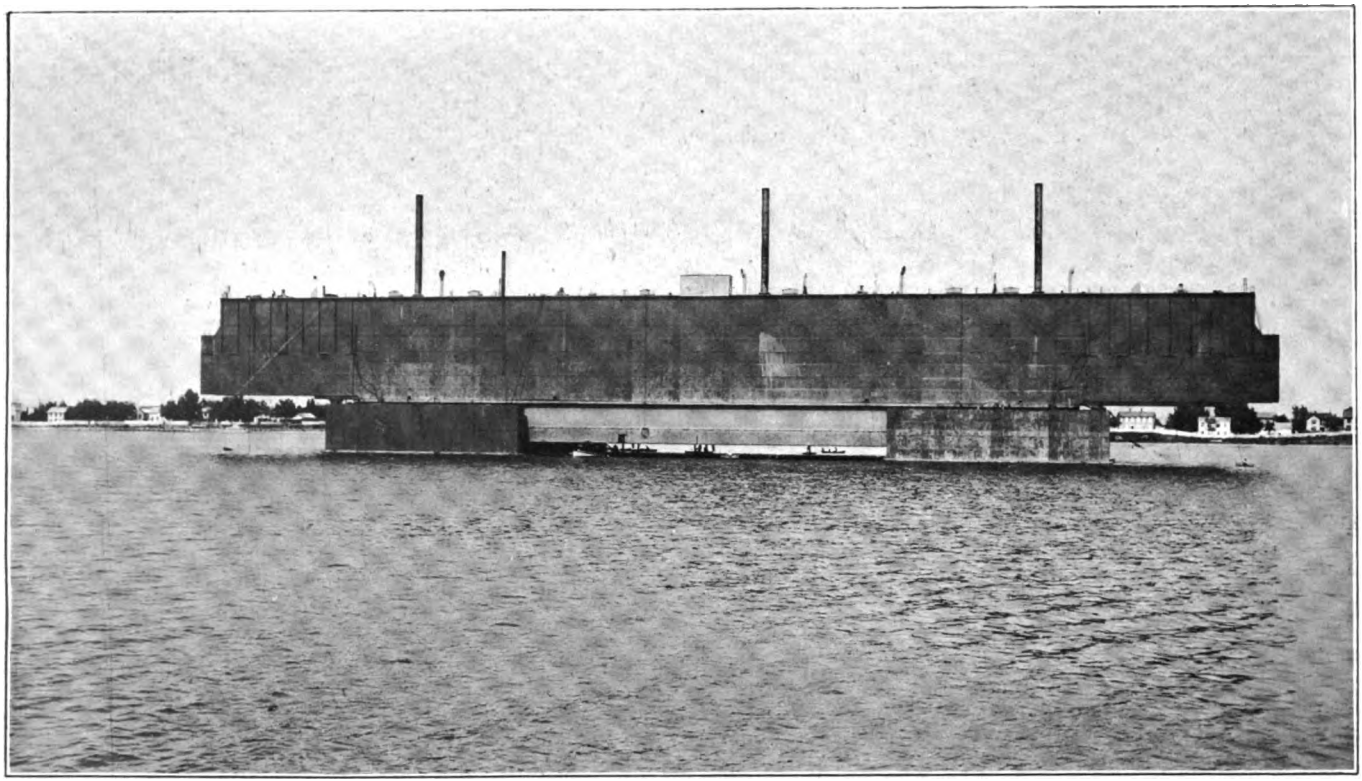

U. S. Floating Dry Dock "Dewey."—Center Pontoon Self-Docked on End Pontoons.

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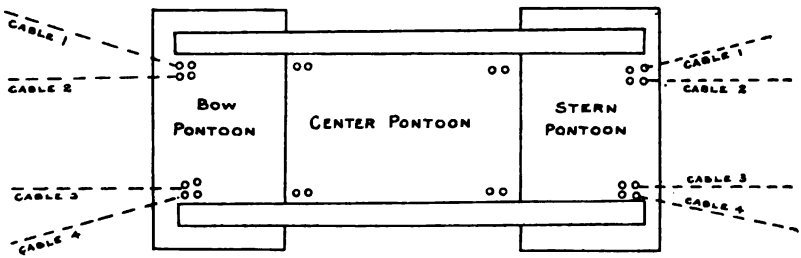
In regard to the desired self-docking qualities of the dock, the specifications provide that it "shall be so designed and arranged as to be readily self-docked without the aid of divers or auxiliary constructions;" that the working stress of any portion of the dock or its connections shall be limited to "15,000 pounds per square inch in self-docking, with a wind pressure of 30 pounds per square foot of exposed surface;" that "when self-docked all under-water portions shall be raised to a height of not less than 5 feet and shall be safely and readily accessible for inspection, painting and repairs;" and, that with the dock at light-draught line, "all self-docking and strain-transmitting connections shall be above water."



Although plans and complete descriptions of the structure have already appeared in the JOURNAL (Vol. XV, page 472) and other technical publications, it may help to a clearer understanding of the operations to give, briefly, a general outline of the design in so far as it affects the self-docking features before attempting to describe the working tests. The dock is built in three sections. The center section, or pontoon, is 316 feet in length, with side walls overhanging 80 feet on either end. Two end pontoons, each 90 feet in length, with low independent side walls, are attached to the center pontoon in such a way that the overhanging side walls of the latter are enclosed between the independent walls of the end pontoons and rest directly on their decks when the sections are connected for docking ships. There are vertical and horizontal connections between the pontoons, the vertical connections consisting of seven elements, each made up of forty-four 2-inch

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bolts; the horizontal, of four elements, one along each edge of the overhanging side walls, each made up of ninety-six 1½-inch bolts. The lower ends of the vertical connections are out of water at light draught.

The self-docking is accomplished in two stages. After removing the connection bolts, the end pontoons are hauled clear of the overhanging walls of the center pontoon, turned so that the direction of their lengths is perpendicular to the axis of the dock, and brought over the deck of the center pontoon and centered on blocking, as in the docking of ships. The center pontoon is then pumped to the desired freeboard by the main pumping engines. The second stage of the operations has for its object the lifting of the center pontoon. The end pontoons are submerged to a depth giving twelve inches clearance between the blocks and the bottom of the center pontoon at light draught, and are then drawn under either end and centered, after which they are pumped to the desired freeboard by means of a separate pumping plant installed in the independent side walls, to which steam is furnished by a flexible hose led from the forward and after boilers.

On Saturday, July 15th, 1905, the work of arranging the blocks of the center pontoon for receiving the two ends was finished, and all vertical connection bolts, together with half of the horizontal bolts, were removed. On Monday, cables 1 and 2 (see accompanying sketch) were slipped from the end pontoon bitts, starboard side, forward, and made fast by hawsers to bitts on the top of the overhanging side walls. On Tuesday, cables 3 and 4, forward, were similarly slipped and belayed on the port side-wall bitts. All horizontal connections were removed and steel drift pins placed in the bolt holes at intervals of about twenty-five feet, with men standing by for each pin. At 12·15 P. M. the flood valves of the forward pontoon were opened, and ten minutes later the horizontal connection diaphragms came gently apart. With the aid of a small tug boat, the end pontoon was hauled clear of the overhanging side walls at 12·53 P. M., and made fast against the starboard side of the center pontoon at 1·35 P. M. Cables 2 and 3 were

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then slipped from the tower-deck bitts and made fast to the forward-deck bitts of the center pontoon with two shots of chain which had previously been detached from cables 1 and 4. An anchor-hoy and tug was used in shifting moorings and handling cables.

On July 19th, cables were slipped from the after pontoon bitts and made fast to the side-wall bitts as was done in the case of the bow pontoon. All horizontal connection bolts were removed and the flood valves opened at 10·20 A. M. The pontoon began to drift out with the tide at 10·49, and was hauled to the port side of the dock and made fast at 12 M. Cables 2 and 3 were then slipped from the tower bitts and made fast to the after center pontoon bitts as for the bow pontoon.

On July 20th, the center pontoon was submerged to an average draught of thirty-three feet, giving a little over twelve inches clearance between the bottoms of end pontoons and the blocking for a draught of nine feet. The end pontoons were then hauled in and centered over the keel blocks. Pumping was commenced at 12·26 P. M., the blocks were awash at 1·25, and the pumps were stopped at 1·40 with a freeboard of eighteen inches, the total time of pumping being one hour and twenty minutes. All mud valves in end pontoons were opened at 1·00 P. M., and the compartments drained with the falling water.

During the next three days the pontoons were held on the blocks and the time was spent in cleaning and repainting such parts of their bottoms as seemed to require it, and in arranging the blocking on the ends for seating the center pontoon. On July 24th the mud valves of the end pontoons were closed at 8·40 A. M., the flood valves were opened at 9·15 and closed with the pontoons afloat at 10·00 A. M. The pontoons were lashed together in tandem and drawn out with the tide, being clear of the dock at 11·31 A. M. Both pontoons were then made fast to the starboard side of the center pontoon. When floated, the end pontoons had an average draught of 4.59 feet, making their joint displacement 3,164 tons.

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The 25th of July was consumed in arranging the blocking, and on the 26th the work of docking the center pontoon was begun. Cables 2 and 3 were slipped and led by hawsers to side-wall bitts, the dock being held in the meanwhile by cables 1 and 4. At 8·30 A. M. the stern pontoon was sunk to sufficient depth to permit its being hauled under the overhangs, and at 11·40 it was in position and ready for sinking to full draught. At 1·05 the flood valves were opened, and at 3·50 P. M. the pontoon had reached an average draught of thirty-one feet, the sinking being carried on very deliberately on account of the possible effect of the strong ebb tide, nearly always to be contended with at this point. After reaching this depth the pontoon was drawn under the center pontoon and centered over the blocks. The pumps were started and continued until the pontoon reached a draught of twenty-eight feet, and the center was firmly seated.

On July 27th the dock was simply held in position and observations made for possible settlement. On July 28th the bow pontoon was drawn into position, sunk to a draught of thirty-one feet, drawn under the center pontoon and, at 2·20 P. M., pumped to a uniform draught with the stern pontoon. Both end pontoons began pumping full speed at 2·20 P. M. and reached a freeboard of eighteen inches at 7 P. M. After holding the dock in this position for four days, the work of undocking was commenced, but, on account of bad weather, this work was discontinued and the end pontoons were held at twenty-six feet draught until August 2d, when the stern pontoon was hauled from under the center and connected in its proper position. Two more days were lost on account of the weather, and the bow pontoon was withdrawn and connected on August 4th. No difficulty was encountered in entering the connection bolts, and each pontoon was sunk, hauled out, pumped to proper draught and connected in twelve hours time.

The total working time consumed in the self-docking tests, exclusive of the time spent on the blocks, is about fifteen days, and, when it is taken into consideration that this work is

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absolutely new, that it was necessary to proceed with caution during the first test, and that the crew was naturally unused to its work, it would seem probable that the time for future self-dockings could easily be reduced to ten or eleven working days.

Numerous devices have been exploited in the past, all claiming more or less novel means to enable floating docks to lift their own hulls for inspection and repairs. Few of these devices appear practical, and still fewer have been actually tried on large structures. The most prominent among the self-docking types is the Clark & Stanfield modification of the Rennie type, represented in this country by the naval docks at Algiers, La., and at Pensacola, Fla. The Algiers dock was successfully self-docked by her builders during the acceptance tests, and required for the operation about forty days. It succeeded in lifting the pontoons to a height of four feet and the side walls to a height of twenty-one inches above water. The Pensacola dock has docked her intermediate pontoons, but is only now, for the first time, attempting to lift the end sections. So far as I am aware, the new Bermuda dock has never been completely docked, nor has the new naval dock at Pola, Austria, the latter being an entirely new design.

The Cavite dock has, so far, proven a complete success, and the self-docking features are particularly satisfactory. In the light of experience with other self-docking types, it would seem that, as regards the saving effected in time and labor, facility of control and simplicity of operation, the self-docking problem has been satisfactorily solved and the greatest objection to the floating dock *per se* completely eliminated.

The self-docking tests were conducted by a Board of which Captain Adolph Marix was senior member, and Naval Constructor W. G. Du Bose and Civil Engineer Leonard M. Cox, members.

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SUGGESTIONS AS TO THE ADAPTABILITY OF A UNIVERSAL STANDARD FLANGE LIST FOR MARINE WORK.

By Luther D. Lovekin, Chief Engineer of the New York Shipbuilding Co., of Camden, N. J., Associate Member.

The subject of flanges for all purposes where pipes are required to be connected by such means is one of the most important details in connection with modern engineering, and one which, without doubt, has received the least attention among designing engineers. Too often does it occur in the drawing rooms of large establishments that when a flange list is to be made out it is given to some young draftsman of very limited experience, with a view, no doubt, of economizing, and when finished we find this new list embodies the mistakes of the past as well as some new features which are liable to add to the already innumerable troubles existing in this line.

With some knowledge of the troubles existing in this line (due to various causes), the writer decided that the time was very close at hand when engineers must give this subject their very best attention, and this will be very apparent to all when we consider for a moment the great tendencies toward the use of higher boiler pressures and their consequent economy. As a result of this, during the year 1899, I took my initiative step in this line, in first inventing a new form of pipe joint, and in order to be certain that I would cover all possible merits in connection with the same, I went at the designing of this flange with as much pleasure as though I were designing an engine. I made all sorts of calculations, such as would apply to this line of work, with a view of placing the metal of the flange in the place where it was most needed, and before I had

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finished my first series of designs and had gathered together all the data I could, I almost concluded that this branch of engineering work had been left to the old and familiar method of "rule of thumb." Be it as it may, I cannot refrain from expressing the view that, while prominent engineers have frequently taken this subject up, they have seldom given the matter serious thought, but, instead, as a rule, when a matter of this kind has been brought to their attention, they have simply asked what the trouble was, and in the case of a flange springing between bolts, or the bolts themselves being weak, they have either said to the draftsman "Put another ⅛-inch on the thickness," or "Put the next size larger bolts in, and I think you will avoid the trouble."

How many engineers have ever taken the trouble to analyze the forces set up in flanges of the various sizes with a view of determining what would be the most suitable shape of a flange, the amount of surface a gasket should have in order to produce a good joint, and, furthermore, to see that whatever flange was used would, as far as practicable, be of the least weight consistent with strength? It seems to me there has been more talk on such items as *multiples of four bolts, being absolutely essential to any standard*, and the thoughts as to how this or that suggestion would affect the valve maker, with his *numerous patterns* on hand, than as to how this whole matter was going to affect the engineer whether at sea or on shore, as well as the owners of our large vessels and industrial works.

There is no doubt in my mind but that the industrial establishments throughout our whole country are suffering from leaky joints in steam pipes, and spending thousands of dollars on various kinds of gaskets or other jointing material in order to remedy defects which are not always due to the kind of material used for such gaskets, but are chiefly due to a lack of strength in the flanges themselves. The writer firmly believes in standards, and appreciates the fact that others before him have made standards, but appreciates the additional fact that no good standard flange list appears to be in general use at the present time for the *higher pressures;* in fact, some of our

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best concerns in New York City today are ordering forged-steel flanges of the same dimensions as the standard cast-iron ones in use at the present time for large power plants. Surely they are not doing this unnecessarily? No, for I have been informed by parties well versed in this line that they are ordering these ponderous steel flanges for safety only, and for this latter cause we cannot help but admire them.

However, we all know that the weakest link in a chain will be the one to first break under great strain, and, therefore, why not adopt some logical reasoning with this subject, just as we do in all other engineering subjects, and then when we have finished our work we can stand on some firm foundation, particularly so after a complete series of tests are made which confirms the theory of our work.

Personally, I see no reason for any variation in the practice of pipe fitting of any character; what we all want is a universal standard, but we will never have this so long as the mechanical engineers of the world adhere to the *multiple of four bolts*, for we all know that with ship work, where every pound of metal wasted means just so many pounds less cargo carried, and, naturally, so much less earning capacity of the vessel. Furthermore, in our naval work, we are compelled to pay a penalty of $500.00 per ton, over and above the weights specified by the Engineering Department.

With such restrictions as these resting upon the designers, builders or owners, it becomes of interest to all to see that there is no unnecessary excess of weight carried on ship-board, such as would necessarily follow were we to attempt the use of the bulky and yet weak flange list known as the *Mechanical Engineers' Standard*, which, in most cases under my supervision, has proven so inefficient in the number of bolts used as to often require the placing of additional bolts between these *multiples of four* after the work has been erected. Personally, I do not believe there is a shipbuilder or marine engineer in the United States who would support such a standard as is used in the general commercial work for pressures above 150 pounds. If this is correct, why, then, should we not all

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unite in one standard, at least for the higher pressures, say from 150 pounds steam pressure and upward? This would enable us all to use valves or fittings of any character for the conveyance of steam, which we all know should be accomplished in the same manner, whether it be aboard ship or in our power houses. There is positively no good reason why we should not have that same factor of safety in our power houses that we have in our ships, and *vice-versa.*

Furthermore, in my opinion, any standard flange list should have the joints or flanges *equal at least to the bursting strength of the pipe itself*, so that we could rest assured that, in the case of water hammers and other shocks of a similar nature we had at least done our best from an engineering standpoint.

The writer having given his best attention to the subject of pipe flanges for several years, and constantly observing the various troubles that were brought to his attention, begs to submit to the marine-engineering world a series of flanges which have been designed very carefully, with a view of overcoming all possible objection, and will be very glad to hear from any of our engineers relative to the same. It might be well to state that my reason for calling particular attention to the marine-engineering world in this direction is due to the lack of any one standard which covers both the Navy Department and the merchant marine.

With this in view, I present herewith a full list of flanges (see sheets Nos. 1, 2 and 3), from 2 inches to 20 inches diameter, designed for 300 pounds working pressure, the same diameter of flange, diameter of pitch circle and number of bolts, however, to be used on all pressures between 100 and 300 pounds, which I have termed "A Proposed Universal Standard Flange List," the intention being to use the list as designed for pressures between 200 and 300 pounds, and then to modify the size of the bolts used for pressures between 100 and 200 pounds on all pipes above 4 inches diameter, so that we would never use less than a $\frac{5}{8}$-inch bolt within the range of sizes given.

It has been my aim to produce a list whereby various methods may be employed for attaching flanges to pipes with-

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out affecting in any manner the principal points of advantage sought for, and in order to fully understand my meaning in this direction I have had prepared sections of 10-inch pipe with their flanges attached as shown.

It will be noticed that I have made a comparison of the single-riveted flange with the pipe flared at the outer end to an angle of 20 degrees. (See sheet No. 4.) I have also shown the double riveting, such as we have used on all our merchant ships, with the pipe caulked inside and outside, similar to that of a boiler-shell joint. (See sheet No. 5.)

Furthermore, I have prepared the double-riveted joint as used by the U. S. Navy, wherein the end of the pipe is pened over into a recess in the face of the flange. (See sheet No. 6.) In all these designs I have kept the section where the bolt passes through the same throughout. Knowing this section to be more in accordance with mechanical principles than the parallel section commonly employed, caused me to adopt this form, and I have had numerous samples made and tested, so as to prove my statements.

Furthermore, I might mention that in order to be certain as to what could be accomplished in the direction of expanded joints, I made several samples of steel pipe 20 inches in diameter and one-half inch thick, and the work done was such that the Committee of Science and Arts of the Franklin Institute awarded me the Elliott Cresson Gold Medal for my work in this new field of engineering. In all cases where expanded joints are made, I am certain that, if necessary, we can burst the pipe without in any way affecting the expanded joint when properly made.

Furthermore, the expanded joint is at least 25 per cent. stronger than the average form of riveted joint, such as shown in the accompanying prints.

It is also absolutely impossible to have a leak between the flange and the pipe when so expanded, for I have made numerous tests wherein I have heated both the pipe and the flange red hot, and then plunged the same in water, and afterwards placed them under a test pressure of 5,000 pounds per square

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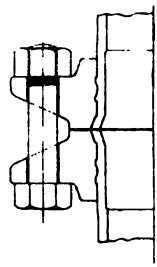
NOTE.-

THIS LIST IS SO ARRANGED THAT FLANGES HAVING A DEPTH OF BOSS AT THE BACK SUITABLE FOR EITHER SINGLE OR DOUBLE RIVETING CAN BE EITHER CONNECTED TO, OR THEY CAN BE USED IN LIEU OF THE EXPANDED PIPE TYPE WHEN SO DESIRED, WITHOUT AFFECTING THE PITCH CIRCLE DIAMETER OF FLANGE, OR THE SHAPE OF THE SECTION.

THEY ARE ALSO SUITABLE FOR CONNECTING ANY ORDINARY CASTING OR VALVE WITHOUT CHANGE OF DIMENSIONS.



NOTE.-

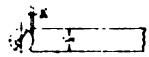
FOR PRESSURES BELOW 175 LBS. THE MALE AND FEMALE JOINT MAY BE OMITTED AS SHOWN ABOVE.

ALL PIPES WITH THEIR FLANGES CONNECTED WILL BE TESTED TO 2½ TIMES THE WORKING PRESSURE

THESE FLANGES ARE TO BE USED ON COPPER, IRON AND STEEL PIPE.

ALL FLANGES TO BE OF GOOD MOULDED IRON OR STEEL OR STEEL CASTINGS.

BOTH FLANGES CAN BE MADE FROM THE SAME PLANS.



SHAPE OF IRON FOR FORMING ALL ABOVE.

SHEET No. 1.

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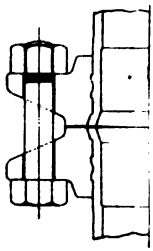
SHEET No. 2.



NOTE:-

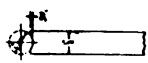
FOR PRESSURES BELOW 175 LBS, THE MALE AND FEMALE JOINT MAY BE SUITED AS SHOWN ABOVE.

ALL PIPES WITH THREE FLANGES CONNECTED WILL BE TESTED TO 2½ TIMES THE WORKING PRESSURE.

THESE FLANGES ARE TO BE USED ON COPPER, IRON AND STEEL PIPE.

ALL FLANGES TO BE OF GOOD MERCHANT IRON OR STEEL, OR STEEL CASTINGS.

BOTH FLANGES CAN BE MADE FROM THE SAME BLANK.



SHAPE OF IRON FOR CUTTING AN ARCHED

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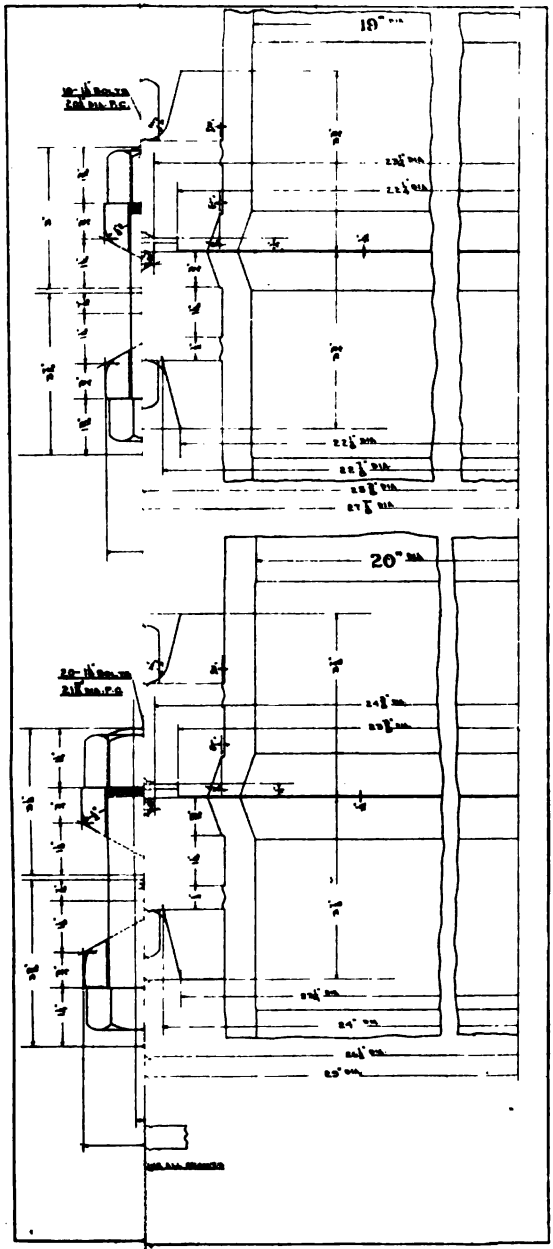

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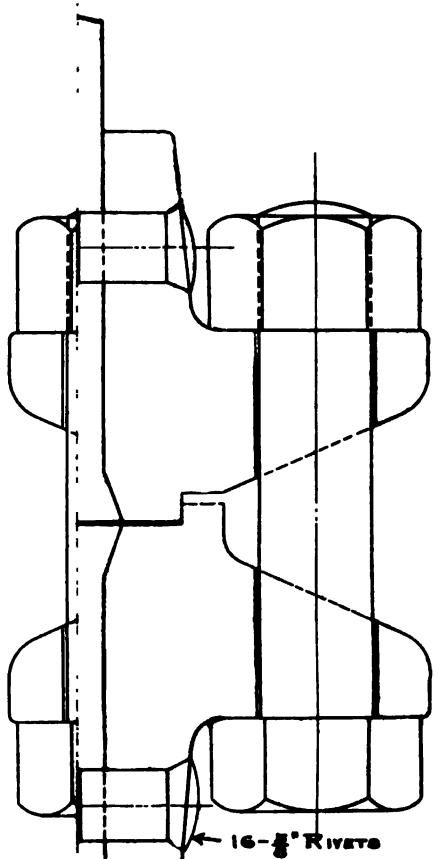

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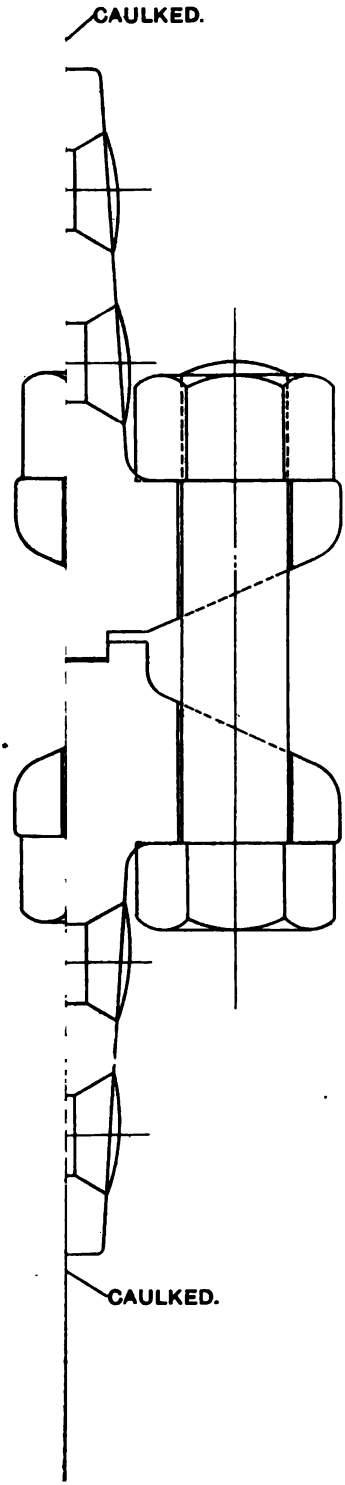

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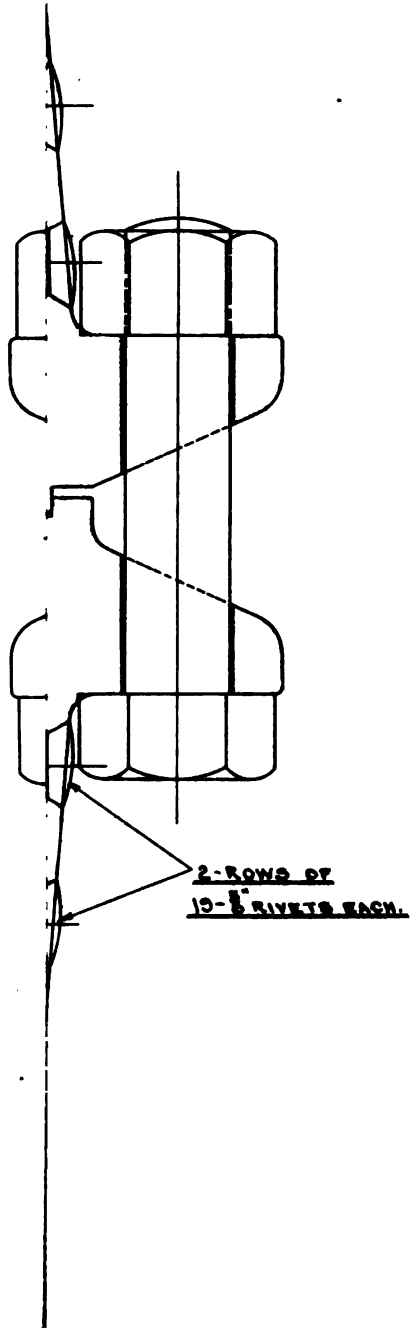

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inch, without any sign of leakage whatever. *This will be apparent to all when it is considered that the ends of the pipes themselves form the joint and any additional clamping effect produced by the bolts has a tendency to pull the pipe closer to the flange, in direct proportion to the angle of flare at the end of pipe, the small shoulder at the end preventing the pipe being forced out of the flange.* As stated previously, my aim has been to produce a flange list whereby any of the methods shown may be employed without affecting the principal points of advantage sought for.

Furthermore, we all realize what a great advantage would be had in a universal flange list, and in order to accomplish this purpose it will require the united efforts of both the shipbuilders, the Navy Department and the Bureau of Commerce and Labor (Steamboat Inspection Service). This latter Bureau I have satisfied, and any of the forms shown herewith meets with their approval, according to the latest rules and regulations (1905 edition).

In order that there should be no one particular form of flange connection considered in this matter I have, as before stated, considered all of the joints used in marine work at the present time, with the exception of the welded flange. My reason for not considering this is on account of the enormous expense necessary to make the same, and also the inability of using a flange welded to the pipe to its greatest advantage, such advantage being a very small diameter of flange and also the ability to use a small pitch circle, due to not having any boss on the back of the flange; *this type of joint could never be employed as a universal standard*, inasmuch as any standard flange list adopted must have the following characteristics, viz:

We should be able to join a pipe to a valve or similar casting, when made of cast iron, the thickness of metal in such valves or castings being always in excess of the thickness of material in the pipe itself; we should also provide for pipes having their flanges rivetted to the pipe. Consequently we must have a pitch circle that will readily conform to the above,

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so that the nuts or heads of the bolts may clear the fillet in the casting. All flanges submitted by me have had these points fully considered, and the advantages sought are as follows:

First. A uniform outside diameter of pipe flange, for pressures between 100 and 300 pounds.

Second. A uniform pitch circle for bolting flanges together.

Third. A sufficient number of bolts to be equal in strength to the bursting pressure of the pipe.

Fourth. A flange section fully as strong as the bolt section in all cases, so designed that the weight of same is reduced to a minimum.

Fifth. To have all bolts entirely outside of the gasket, thereby avoiding troublesome leakage through bolt holes, such as is often encountered.

Sixth. That the areas of the gaskets between flanges may be subjected to a uniform pressure per square inch in all cases by means of the number of bolts used, said total pressure on gaskets being in excess of the tendency to pull joints apart (due to the internal pressure in the pipe) without subjecting the bolts to a stress of over 5,000 pounds per square inch under working pressure.

This feature of the gasket area is a very important one, and the design submitted requires the mimimum amount of material that could be desired for such purpose, thus securing a great saving to both the builder and owners in the supply and renewal of these gaskets.

Seventh. The flanges, as shown, may be used for either copper or steel pipe, and can be made of steel or iron for copper pipe when not subjected to corrosion (such as pipes in the bilges of ships or in any other locality where galvanic action plays an important part), therefore *eliminating the practice of brazing the copper pipe to the flange*, which, from results shown, is positively not a necessity, but rather a source of danger. Where necessary or advisable to use brazing metal or bronze flanges, they may be of the same sectional area of the steel flanges shown, inasmuch as these steel flanges have been designed for steel pipe, and the same sections may be made of brazing metal

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for copper pipe, and bear practically the same relation as the steel flanges for the steel pipe.

The third and fourth paragraphs are the ones I have given great consideration, and are the basis of all my work. I am not a believer in that practice of engineering which permits the use of varying factors of safety for the same structure when subjected to all kinds of shocks, such as water hammers, etc. We should certainly have the pipe with a certain factor of satety and then make the flange connection with approximately the same factor. In all of my work, as will be clearly shown in Fig. No. 7, I have taken the bursting strength of the pipe as the basis for all calculations relating to these joints. A full set of calculations appear on pages following, which clearly indicates the manner in which each and every one of these flanges have been designed.

It will be noted that for pressures above 175 pounds I have used the male and female type of joint; for pressures below 175 pounds a plain joint is used, such as shown in the drawings of proposed universal flange list. While, as stated before, any of our well-known methods may be employed in connection with this universal standard flange list, I would state that the advantages of the expanded pipes as shown are as follows:

No heating, no brazing, no riveting, *no leakage such as is often encountered through rivet holes or between pipe and the inside of flange*, no crystallization. Joint may be designed to exceed the strength of the pipe. By the omission of rivet holes the pipe is about 25 per cent. stronger than riveted pipe.

Furthermore, I have made many samples wherein from actual tests the pipe showed an increase in strength due to the cold rolling it received of about 15 per cent.

In conclusion, I would state that with a flange constructed on this principle the tightness of joints at the gasket will not be affected by the pitch of the bolts, inasmuch as the great depth of flange over the gasket is not susceptible of any deflection whatever, and it is therefore impossible to blow a gasket out, due to the deficiencies such as occur in the older forms

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of flange connections. All that is needed in a design of the character shown is a sufficient number of bolts to enable the flanges to be drawn together to such an extent as to produce a pressure on the gasket in excess of the tendency to pull apart, and a section of flange that will not be unduly strained under such pressure. The depth of the flange through the bore must be sufficient for either single or double riveting or expanding, and the depth of bores in the expanded joints shown herewith are the results of numerous tests made by me, and are what I consider the least possible for safety.

I enclose herewith a table (marked Fig. No. 8) showing all the various stresses throughout, of this proposed universal flange list, which gives the various proportions throughout in a condensed form, and I trust that this matter will be considered of sufficient importance to give the same thorough consideration, knowing, as I do, that this is a matter which has been under consideration for some time, and believing that a flange list for the Navy Department should not differ in any way from a flange list for the merchant marine, causes me to introduce this subject in the manner I have done.

A comparison of these flanges with those in use by the U. S. Navy Department will show very clearly the great saving in weight over the above-mentioned flanges, and in addition will also convince one of their superiority over the former standard flanges as regards strength. (See list of comparative weights, Tables No. 9 and 10.)

[NOTE.—A full and complete discussion is invited on this subject, and it is hoped that the various shipbuilders and engineers will give this matter their consideration on account of its great importance, addressing their remarks to the Secretary-Treasurer.]

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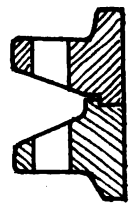
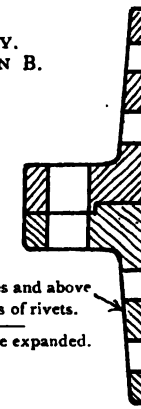
Table IX.

COMPARISON OF WEIGHTS BETWEEN "LOVEKIN UNIVERSAL" PROPOSED LIST (COLUMN A) AND THE U. S. NAVY DEPARTMENT, BUREAU OF STEAM ENGINEERING, STANDARD. WEIGHT OF ONE PAIR, MALE AND FEMALE, ALL CALCULATED OR SCALED; NO DRILLING, BUT BORED AND FACED TO WITHIN ONE-SIXTEENTH INCH OF DRAWING DIMENSIONS.

Diameter.	Lovekin. Column A. 	Navy. Column B. 
Inches.	*Pounds.*	*Pounds.*
2	12.5*	As they come from the die. 12.5†
2½	14.6*	17.†
3	17.3*	25.†
3½	20.8*	28.†
4	23.9*	34.†
5	35.6*	44.†
6	45.1*	72.8*
7	58.8*	83.7*
8	73.1*	103.3*
10	105.7*	135.4*
12	144.7*	195.*
15	221. *	275.*

* Estimated. † Scaled.

NOTE.—The Lovekin flange is very much stronger than the Navy type as shown, owing to the thickness of flange at the bolt section. Flanges are less difficult to make, and thus a great saving is obtained in price, owing to the omission of the great depth of boss.

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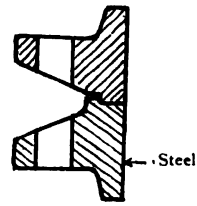
Table X.

COMPARISON OF WEIGHTS BETWEEN "LOVEKIN UNIVERSAL" PROPOSED LIST (COLUMN A) AND THE FLANGES USED ON THE *WASHINGTON* AND *KANSAS* (COLUMN B), FOR 300 POUNDS GAUGE PRESSURE.

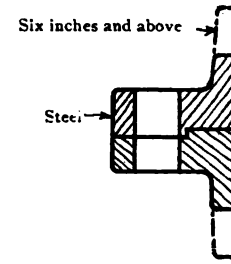
Weight of one pair (male and female) faced and straight bored to within one-sixteenth inch of drawing dimensions, calculated or scaled without any bolt holes or rivet holes being considered.

COLUMN A.



COLUMN B.

On six inches and over collar is lengthened and a single row of rivets used.



DIAMETER.		COLUMN A.	COLUMN B.
Inches.		*Pounds*	*Pounds.*
2	As they come from the die.	12.5*	12.5†
2½		14.6*	17.†
3		17.3*	25.†
3½		20.8*	28.†
4		23.9*	34.†
5		35.6*	44.†
6		45.1*	57.†
7		58.8*	68.*
8		73.1*	86.†
10		105.7*	115.8*
12		144.7*	166.†
15		221. *	237.*

* Estimated. † Scaled.

NOTE.—The Lovekin flange is very much stronger than the ones used on the *Washington* and *Kansas*, owing to the thickness of flange at the bolt section.

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CALCULATIONS FOR A 12-INCH FLANGE.

See Figure No. 7.

Inside diameter of pipe, nominal, 12 inches.

$$T = \frac{P \times D}{10{,}000} + \tfrac{1}{8} \text{ inch, U. S. Navy rule.}$$

$P = 265$ pounds per square inch.

$D = 12$ inches.

$$T = \frac{265 \times 12}{10{,}000} + \tfrac{1}{8} \text{ inch} = .443, \text{ say } \frac{29}{64} = .4503.$$

Bursting pressure $P = \frac{2TS}{D}$.

Assume $S = 50{,}000$ ultimate;

Hence, $P = \frac{2 \times \frac{29}{64} \times 50{,}000}{12} = 3{,}780$ pounds per square inch;

Factor of safety for 300 pounds $= \frac{3{,}780}{300} = 12.6$.

Use 16 bolts 1 inch diameter. Area at root of thread $= .55$ square inches.

Assume that the end pull, corresponding to the bursting pressure, will be over the area of the pipe to the inside diameter of the gasket.

Area for $12\frac{1}{2}$ inches diameter $= 122.72$ square inches.

Pull on bolts $= 122.72 \times 3{,}780 = 463{,}800$ pounds.

Pull on each bolt $= \frac{463{,}800}{16} = 29{,}100$ pounds.

Stress per square inch at root of thread $= \frac{29{,}100}{.55} = 53{,}000$ pounds.

Outside diameter of gasket $14\frac{3}{8}$ inches $= 162.3$ square inches, area;

Inside diameter of gasket $12\frac{1}{2}$ inches $= 122.72$ square inches, area.

Area of gasket $= 39.58$ square inches.

When the joint is tightened up, assume that it is compressed to a pressure of 1,000 pounds per square inch, as this may be necessary to imprint the flange faces thoroughly. The resultant stress on the bolts would be $\frac{39.58 \times 1{,}000}{16 \times .55} =$ 4,500 pounds per square inch.

Under working conditions, at 300 pounds pressure in the pipe, assume that this same pressure is maintained over the gasket area, so there can be no chance for the steam to enter between the faces in contact, and thus invite a leak.

Hence, working stress on bolts $= \frac{162.3 \times 300}{16 \times .55} = 5{,}500$ pounds per square inch.

CALCULATIONS FOR STRENGTH OF FLANGE ON THE SECTION *AB* TO RESIST THE MOMENT OF THE BOLT WHEN THE PIPE IS SUBJECT TO A BURSTING PRESSURE.

Assume that each bolt acts on its own proportionate part of the flange; that is, the flange is divided into sixteen beams, fixed at one end and carrying the pressure of the bolt at the other.

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F = pressure of one bolt due to the bursting pressure in the pipe = 29,100.
L = distance of application from neutral axis = $1\frac{1}{32}$ inches.
Hence, moment = $F \times L$.
Moment of resistance for this beam = $\frac{sbh^2}{6}$.
h = depth of beam (length of AB) = $1\frac{11}{16}$ inches.
s = maximum fiber stress in the section.
b = breadth of beam measured along the circumference of neutral axis = 2.88 inches.

$$F \times L = \frac{sbh^2}{6}.$$

Hence, $S = \frac{6FL}{bh^2} = \frac{6 \times 29{,}100 \times 1\frac{1}{32} \text{ inches}}{2.88 \times (1\frac{11}{16})^2} = 22{,}000$ pounds per square inch.

Thus the flange cannot be bent by the bolts when the pipe is subject to its bursting pressure, on account of our keeping the stress below the elastic limit of the material in the flange.

Stresses on Grooves When the Pipe is Subject to Bursting Pressure.

Area of shoulder, $\frac{3}{64}$ inch deep.
$13\frac{1}{16}$ inches outside diameter = 133.9 square inches area.
$12\frac{31}{32}$ inches inside diameter = 131.9 square inches area.
= 2. square inches net area.

The projected area of the curved shoulder is also equivalent to 2 square inches area; hence the bearing area of the grooves = 2 × 2 = 4 square inches.

The projected area of the flare is the most effective bearing surface against end pull on the pipe.

$13\frac{1}{2}$ inches outside diameter = 143.1 square inches area.
$12\frac{31}{32}$ inches inside diameter = 131.9 square inches area.
= 11.2 square inches net area.
4
Hence the total bearing area = 15.2 square inches.

Crushing Stress Due to Bursting Pressure in the Pipe.

$$\frac{122.7 \times 3{,}780}{15.2} = 30{,}500 \text{ pounds per square inch.*}$$

Depth of shearing area = depth of grooves + depth of flare = $\frac{1}{2}$ inch + $\frac{9}{16}$ inch = $1\frac{1}{16}$ inch.
Circumference of shearing area for $12\frac{31}{32}$ inches diameter = 40.8 inches.
Shearing area = 40.8 × $1\frac{1}{16}$ inches = 43.4 square inches.

Shearing stress due to *bursting pressure* = $\frac{122.7 \times 3{,}780}{43.4}$ = 10,700 pounds per square inch.†

* This stress could be at least 50,000 pounds.
† This stress could be at least 40,000 pounds.

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IMPORTANT NOTE.

The principal bearing surface in this form of joint is the projected area of the flared portion at the end of the pipe. The grooves in the bore of the flange, however, have a very important function with this type of joint, serving to retain the flange in a positive position with relation to the end of the pipe, and also partially relieving the shearing and crushing stresses at the ends.

The complete filling of these grooves is not essential to the effectiveness of this type of joint.

See cut No. 7A, showing diagram of the resolution of forces, attached herewith.

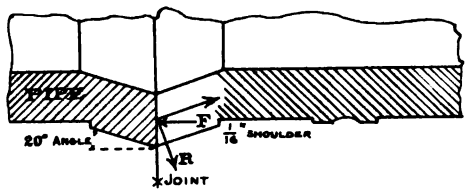

Fig. 7a.

Diagram of Resolution of Forces in Lovekin Method of Making Flange Joints.

Note the outward reaction of flared pipe due to end pressure.

$R =$ reaction.
$F =$ end pressure.
$R = F \sin 20°$.
$R = .342\ F$.

The coupling also tends to close in on the flare of the pipe, due to the moment of the bolts. This maintains the close contact between the pipe and the flange.

The dimensions of pipe flanges for universal standard follow in table, Fig. No. 8.

The method of calculating the pull imposed on the bolts by the bursting pressure in the pipe may be open to question. It can be argued that it is better to consider the pressure over the whole area of the pipe and gasket, instead of taking it to the inside diameter of gasket only.

In accordance with this argument that the pressure is distributed over the whole area, it will be well to give a column of figures showing the working stress on the bolts due to 300 pounds pressure over the entire surface; the table herewith appended gives such.

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Stress in Bolts due to 300 Pounds Pressure Over the Entire Area of Pipe and Gasket.

Diameter of pipe.	Stress in bolts.	Diameter of pipe.	Stress in bolts.	Diameter of pipe.	Stress in bolts.
Inches.	*Pounds.*	*Inches*	*Pounds.*	*Inches.*	*Pounds.*
2	2,850	7	4,980	12	5,540
2½	2,550	7½	4,100	13	5,720
3	3,510	8	4,560	14	5,220
3½	3,460	8½	5,150	15	5,940
4	4,210	9	4,420	16	6,050
4½	3,510	9½	4,820	17	5,280
5	4,120	10	5,250	18	5,350
5½	4,780	10½	4,970	19	5,930
6	4,540	11	5,360	20	6,000
6½	4,870	11½	5,150		

All bolts to be of either good mild steel or the best grade of wrought iron. No special material required in any case.

Steel of 60,000 tensile strength, iron of 45,000 tensile strength have ample safety in all cases.

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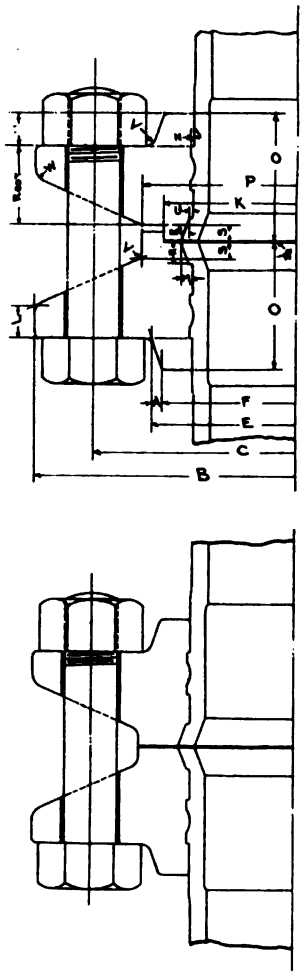
FIG. No. 8.



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EXPERIENCES WITH NICLAUSSE BOILERS ON THE U. S. S. *NEVADA.**

By Lieutenant A. M. Cook, U. S. N., Senior Engineer Officer, Member.

My experiences with the Niclausse boilers on the U. S. S. *Nevada* have been so commonplace that it may seem uninteresting to the searcher after something new in marine water-tube boilers, yet I am tempted to recount some of these experiences.

The *Nevada* was placed in commission at Portsmouth, N. H., March 5, 1903, with C boiler in use for auxiliaries. I was informed by the contractor, the Bath Iron Works, that each boiler had, at that time, been in use about seventy days. Since commissioning, the four boilers have been in use for all purposes, to June 1, 1905, as follows:

Boiler A, 291 days, 19 hours.
B, 270 days, 14 hours.
C, 276 days, 19 hours.
D, 230 days, 1 hour.

On March 6, 1903, two days after commissioning, the draft failed, and the use of the steam lance for cleaning soot off the tubes gave only temporary relief; matters got worse until March 12th, when it became necessary to stop all auxiliaries except the feed pump in order to increase the steam pressure. On that day the forced-draft blowers were started and run for a short period, with the idea of blowing the soot off the tubes; and for several days thereafter no further trouble with the draft was experienced. It was found at that time that the draft would fail after any boiler had been in use for three or four days, and to avoid this it became routine to run both

* For a description of the U. S. S. *Nevada* see Vol. XV, page 105.

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blowers at full speed for a short time each morning watch, using the lance about once a week. The soot settled on the horizontal baffles and the lance would not clear the heating surface; the use of the blowers, however, was effective and disposed of the soot by way of the smoke pipe.

In chronological order, the next thing that caused some anxiety was the attempt to remove the circulating tubes for examination, when it was found that the plugs had been set up too hard. It is quite probable that the mechanics responsible for this condition had had no other experience with the boilers than that of erecting them as new work. The plugs are made with a double thread on a taper, and after having been set up too hard, a little heat did the rest, and it was difficult to remove them; their removal was finally accomplished by patience, the use of extra tools and a wrench for clamping the generating tubes in their seats while unscrewing the plugs. In replacing these plugs it is necessary to coat the threads in with a lubricant which will not burn hard, to start the plugs in carefully, and then screw them home with a moderate strain. A 14-inch wrench with a strain of about 50 pounds is sufficient.

The next trouble was with grease in the boilers. When the ship was placed in commission there was no filtering material in the feed tanks. My attention being devoted to more pressing matters, this important requisite for the care of the boilers had been neglected. Loofa, or rag plant, has since been used with good results, and grease has caused no anxiety. During this experience without filtering material, it was found that the grease was confined almost entirely to the first three headers next the feed inlet, most of it being deposited on the inside of the circulating tubes of these headers. The feed water, in passing first through these headers, as it seems to do, deposited nearly all the grease. The tubes of the adjoining headers were found to have only a trace of grease. On the opposite side of the boiler, away from the feed inlet, no trace of grease was ever found, and, in consequence, it was necessary to examine and clean the tubes near the feed inlet oftener

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than the others. The character of the circulation is, undoubtedly, the cause of finding all the grease in a few headers near the feed inlet and the salt scale in the headers farthest away. The water enters at one side of the boiler and apparently circulates through nearly all the headers in turn until it reaches the other side, at which point the concentration of the water is higher than anywhere else; therefore, the salinometer connection should be at the end of the drum farthest from the feed inlet, instead of in the center, as generally placed.

At the first general cleaning of the boilers, scale was found in all of them. Since then none has been found except in one or two instances, where the circulating tubes had become defective. The boilers had the same feed water, care and attention before as they did after this scale was found, and I am of the opinion that the scale was in the tubes when the ship was delivered to the Government. This opinion is strengthened by a later experience, when unusual and urgent circumstances, combined with a leak in the condenser, caused one boiler to be used for about fifteen hours, with a concentration equal to that of ordinary sea water, and an examination at the first opportunity showed no scale at all in the tubes. During the time that this boiler was in use with salt water, we were burning about fifteen pounds of coal per hour per square foot of grate, and there was not the slightest evidence of any priming. This was a revelation to me, for my torpedo-boat experience led me to believe that such a thing as no priming when using salt feed could not occur with any water-tube boilers. The scale found at the first general cleaning was removed with the turbine cleaner and scrapers, some of which were made on board, having two edges, shaped to the curve of the tube and held out by the spring of the fork on which they are formed. After the scale, grease, or any other kind of dirt, has been loosened in the tube it is removed by a steam jet. This jet is produced by a pipe, somewhat longer than the tube, connected with a steam hose. The end of this pipe is closed, and there are small holes near the end, drilled so as to direct the jet backwards at an angle of about 45 degrees. With the steam turned

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on, this jet will remove all water and loose dirt from the tube. Most of the water in the large tubes is usually removed by a syphon before using the jet, on account of the large amount that would be blown into the fireroom.

After the ship had been cruising for some time in squadron, it was found that some of the malleable iron lanterns of the lower generating tubes were broken, as shown in Figure No. 1; the first one discovered to be in this condition had developed a slight leak around the outer cone joint; an occasional broken one was found in this way, but the greater number were not discovered until the circulating tubes were removed for the routine examination. Requisition was made for new lanterns. In the meanwhile boiler D was in use for some weeks, during which time it was known that several of the lanterns were broken. Some of these had leaked when they first broke, or, rather, when they first came to our notice; but when pressure was off the boiler, they were put straight on their seats and the dog adjusted reasonably tight against the lantern plug; this always stopped the leak. When the boiler was taken in hand for the renewal of these defective lanterns, it was found that not fewer than eleven were broken in the lowest row, out of a total of nineteen in that row. A few others broke in this boiler afterward, but it is probable that these had already been cracked. This trouble is believed to have been due to the material of which the lanterns were made and to the usage to which they were subjected; they are malleable castings, and the usage referred to was the practice of opening the fire doors to control the steam, which practice was of frequent occurrence while in squadron until the trouble of cracked lanterns was understood; since the opening of fire doors to control steam has been stopped, very few lanterns have broken. I believe that one of the old style lanterns is occasionally broken even now while cleaning fires. When the fire door is opened wide, with a hot fire, the great inrush of cold air striking the tubes of the bottom row contracts the lower side of them to a considerable extent, and very much more quickly than it does the top sides of the same tubes; this causes the tube to bend up in the

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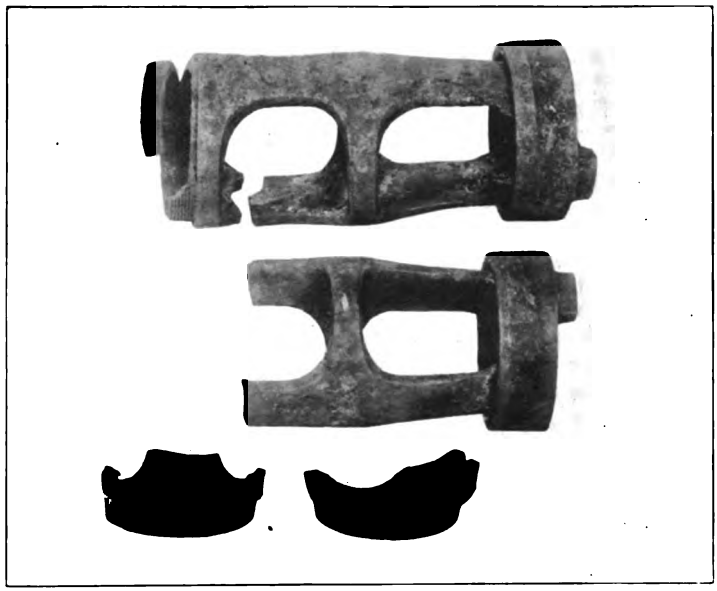

Fig. 1.—Broken Lanterns.

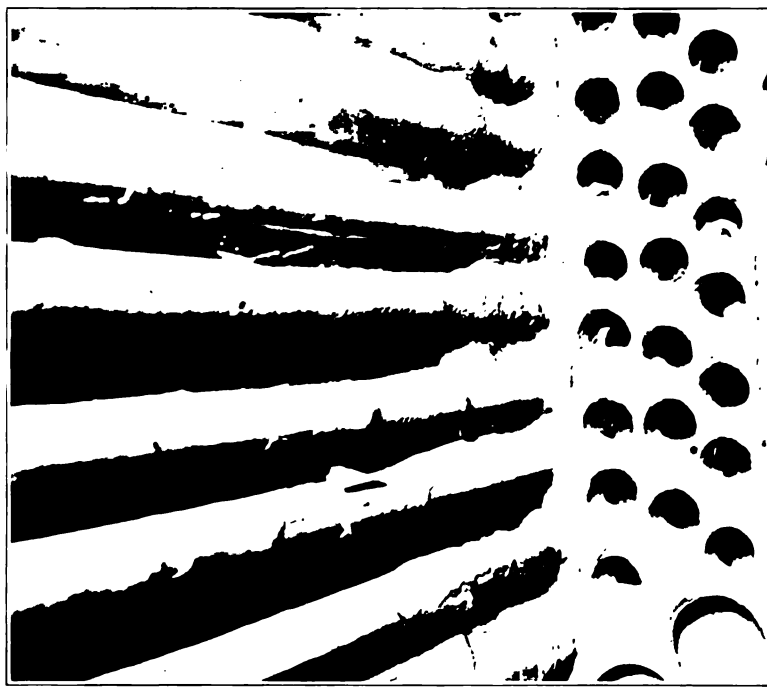

Fig. 2.—Tubes Before Cleaning.

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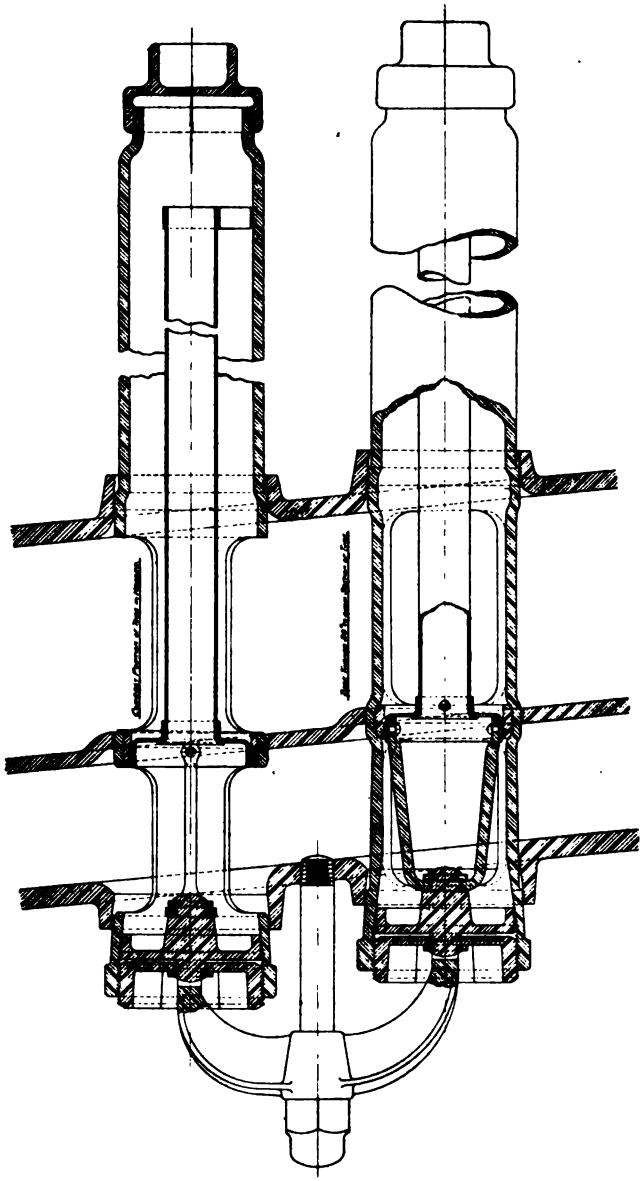

Fig. 3.—New Type of Lanterns.

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middle in the form of an arch throughout its length; at the same time the lantern, being fast in the header, remains in its original position and causes what I am pleased to call a "kink" in the lantern. Either a frequent repetition of this action, or some especially violent one, causes the fracture. An inspection of the furnace not infrequently shows tubes bent up as described; in every case, however, these same tubes will be found to be straight at a later date, and I do not believe that this bending causes any other damage than the fracture of the lanterns. This trouble cannot occur with lanterns made according to present practice. The bending of tubes, however, shows the great strain to which straight tubes, fast to both ends, must be subjected in water-tube boilers with tubes so secured.

The manufacturers of these boilers have long since stopped making lanterns of the pattern originally supplied to this ship. A few of the new style ones have been in use in the *Nevada's* boilers for nearly a year with entire satisfaction. The new lantern is made in one piece, with the tube as shown on sectional elevation of the headers and tubes. (See Fig. 3.) The tubes are received by the manufacturer from the mill of one diameter throughout; the header ends of tubes are upset to compensate for the boring and threading on the inside; they are then expanded at the points where the joints are made on the fire side of the header, and where they pass through the header diaphragm; the loose ends are then swedged and the tubes are passed on to the machine shop, where they are machined, the lantern portion of the tube being milled out of the solid. The improved tubes referred to as having been used on the *Nevada* depended upon the fit of the plug thread for tightness; on the tubes shown, the threaded plug carries a swiveled cone ahead of it, this cone making the joint on the inside of the tube and relieving the thread of the necessity of being steam and water tight. This is a further improvement, and the substitution of a cone for a screw joint and should eliminate any sticking of these plugs. With this construction, which is but an extension of the cone principle for all joints

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in this boiler, I am informed they will test tight at 500 pounds when screwed in with one hand and a 6-inch wrench. This construction does not eliminate the effect of expansion and contraction on the tubes, but its flexibility compensates for these strains, and, as stated, one year's service has not developed any fault.

In putting in new tubes it has happened that the tube did not fit properly in both joints of the header. A reamer is supplied with the boilers for the purpose of reaming out the seats in the header. This has not been used, the method followed being to lap off the cone of the tube which is too large. This latter method is used for two reasons: one, because it is the easiest way to correct the fault; the other, and the controlling reason, being that there is as much liability, if not more, that the fault is in the tube rather than in the header. If the header is changed, the next tube may not fit it, while if the tube is changed, it can always be put back in the same place, and whenever it becomes necessary to put in another tube the header will still be in its original condition.

I have gone into the troubles on board the *Nevada* quite extensively, believing this to be the best means of explaining the way in which they were overcome, and the necessity for precautions against their recurrence. The means taken may not have been the best in every case, but they have not been radically wrong. No boiler of this ship has ever had to be taken off at sea or in port on account of any defect. This is a very good showing for the design and workmanship.

A recital of the routine in caring for the boilers may now be of interest. The circulating tubes of the headers near the feed inlet and those of the lowest row of tubes across the boiler are removed for examination after the boiler has been in use from twelve hundred to fifteen hundred hours, these being cleaned at the same time, when necessary. A few other tubes, selected at random, are opened for inspection, a record being kept so that different ones will be taken each time. If there is any fault general in the whole boiler it will be developed by this means, while any purely local trouble might

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be missed. To avoid this, all the circulating tubes were removed from all the boilers during the second year in commission. Two of the small tubes in the middle of boiler A were found to have defective circulating tubes; one of them had been partly filled with a deposit of scale and had to be renewed, the other was not injured sufficiently to require renewal.

I believe that it would be good policy to make a thorough inspection of all the tubes in a boiler at least once in each year, as while after the damage is done the cause will be found without this work, yet such inspection may prevent possible damage. This thorough examination of all the tubes in one of these boilers requires only a couple of days under ordinary conditions, and in a ship where all boilers are not constantly in use at sea the boilermaker can work at sea as well as in port, and there will be no trouble in finding the opportunity for this work.

When a boiler is to be out of use for more than a very few days it is always entirely filled with water. The water is tested for acidity, and enough alkali is added, if necessary, to be certain that the water is not acid.

Precautions were taken from the first to see that the gauge glasses were always in proper working order; they were blown through every watch. The water end of the glass is connected with the bottom of the drum by a U-shaped pipe, which is fitted with a cock at the bottom of the U. This is always blown before blowing the glass, and it is found that considerable sediment collects at this point. By this means the dirt is prevented from being raised into the smaller passages leading to the glass.

The boilers were originally installed with Niclausse automatic feed regulators. I have no doubt that these will work satisfactorily so long as they work at all, but they were never given a trial after the ship was put in commission. As soon as I took charge of the department, on March 5, 1903, they were all secured in the wide-open position; afterwards they were all removed. They are not necessary with these nor with any water-tube boiler in use in the U. S. Navy. The

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regulation and control of the water level in the Niclausse boiler is not difficult in the least, under ordinary circumstances; one water tender having cared for four boilers when steaming at full power with forced draft. This has been done on the *Nevada* for four consecutive hours while burning 30 pounds of coal per hour per square foot of grate. I am firm in the belief that the use of automatic feed regulators is but an invitation for negligence and disaster. If they are in use, the water tender will naturally become negligent and not watch the water glass; when they cease to work is most probably the time when confidence in them will prevent the water tender from noticing that they are out of order. The water tender must be on duty, whether the automatic regulator is in use or not; if it would allow us to dispense with this valuable petty officer I might be converted to its use, but it does not do so.

There is always some grease in the feed water, and some dirt, to say nothing of salt, in spite of the filter and any other precautions that may be taken. It has been the routine to blow the boilers a short, hard blow twice daily; in the morning at 8 o'clock the surface blow is used, after being sure that the water in the boiler is at the level of a mark on the gauge such that the scum pan is just at the water level; in the evening, at 8 o'clock, the bottom blow is used, alternating on even and odd days with the blows on opposite sides of the boiler.

It had been found that the after boilers were less efficient than the forward boilers, and, in an effort to overcome this difference, if possible, it was decided to change the baffling in C boiler, one of the after ones, believing that the horizontal baffles were holding the soot above them in considerable quantity, and I thought that this accumulation would be reduced by removing an occasional baffle tube to allow the gases to go up through the baffling at intermediate points. To change the baffling it was necessary to remove a few generating tubes, and it was found that all the tubes thus removed were covered outside with a thick, hard scale, of a non-conducting material, and that

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this scale was very hard to get off. Its removal required the use of steel scrapers and a considerable application of "elbow grease." This scale was from one-thirty-second to one-sixteenth of an inch in thickness. I believed that the cause of the lack of efficiency was located, and that it would be solved by removing all the tubes of this boiler and cleaning the fire side of them. Owing to the lack of causes requiring the removal of many tubes and the consequent inexperience of the boilermaker in that work, this was considered to be a more difficult task than it eventually proved to be. Advantage was taken of a stay in port to begin this work, and it was found that the conditions described were the same throughout that boiler. The work progressed very satisfactorily to completion. In view of the condition of C boiler, I determined to undertake this work on all boilers at the first opportunity. Boiler A, a forward one, was next taken in hand. It was at once seen that the condition of the fire side of the tubes of boiler A was entirely different from that which had existed in C boiler. There was a considerable amount of ordinary soot on the tubes and baffles, but it was of such character that a good blow from one's mouth was sufficient to remove it, leaving the tube perfectly clean. There was not a single tube in this boiler which had scale on it like that which had been found on every tube in C boiler. I am still looking for a satisfactory explanation of this peculiar condition found in boiler C.

Fig. 2 shows the condition usually found before cleaning the tubes. This dirt disappears entirely after blowing with the steam lance.

Boiler D was next examined. This is the other after boiler. Before taking up the work in A boiler, the tubes had not had the usual cleaning with the lance after the last steaming. Before starting D boiler I had one side cleaned in this way, leaving the other side in a condition supposed to approximate that which had existed in A boiler. It was found that the side not cleaned was very similar in condition and amount of dirt to what had been found in A boiler. There was no

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hard scale such as had been found in C. On the side that had been cleaned with the lance the tubes were all clean and in perfect condition.

There was, however, an explanation for the fact that D boiler, notwithstanding its freedom of scale on tubes, had shown no greater efficiency than C. When the boilers were first installed, a different system of baffling had been put in. This had been changed before the trial trip, but many of the old baffles had not been removed from boiler D. It is believed that these extra baffles among the generating tubes of the boiler had caused the soot to collect very quickly after the boiler was started, and had thus had a bad effect on its efficiency.

To illustrate the facility with which the Niclausse boiler can be thoroughly overhauled, or re-tubed in emergency, I want to detail the work done on boiler D at Annapolis. The boiler was taken in hand at 10·00 A. M. Wednesday, May 24th, 1905; about four hundred and thirty circulating and generating tubes were taken out, cleaned where necessary both inside and outside, and all replaced; those not removed were cleaned in place; the boiler was ready to test at the end of working hours on Monday, May 29th. The test made on Tuesday morning, May 30th, showed that every tube was absolutely tight at 240 pounds pressure with cold water. The time taken for this work, not counting the filling of the boiler for testing, was about forty working hours. The test on A boiler after the work on it had been done was perfect. The test on C boiler, the first one undertaken, showed only one tube leaking slightly, and that was made tight in a few minutes. The men employed on this work were one boilermaker and one water tender, with such help as could be used to advantage. About half the time the two men mentioned were alone on the job. The help was used principally for cleaning the tubes.

From the experience gained in this work I am free to say that in case of emergency, such as might occur in action, where a boiler had been wholly disabled, that the working force on

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board could completely re-tube one boiler and have it in commission again in three days. This would allow time for renewing headers as well as tubes, in case any of the headers should have been damaged. It would not be necessary to call for any help from shore, or any special mechanics, or any tools other than those habitually in use with the boilers. The tubes can be removed in this boiler, cleaned and examined externally and internally, and the same tubes replaced, giving one the assurance that every tube in the boiler is properly cared for and sound. This, together with the comfortable feeling that one can depend on the continued service of the boilers, without the probability of failure when it is most important that they be kept going at full power, are strong and effective arguments in favor of this boiler.

I desire to say a word which has been repeated by every engineer who has had experience with Niclausse boilers and with whom I have talked, that is: "Before having had practical experience and been in contact with them, I looked questioningly at the multitude of cone joints. I am now convinced that they are one of the best points in the design of the boiler."

The brick work and casings of the boilers have not suffered to any appreciable extent. It has been necessary to rebuild portions of the side walls, but no metal work on sides or back has been necessary; the casing is as tight now as ever, and is not warped. The only trouble in this respect was with the door liners, and this was confined to the cast-iron arch. After a very little use with anything more than easy firing, these arches began to droop, interfering seriously with the firing. While at the Norfolk yard last spring the design of the arch was changed; this also involved some change in the lintel which helps to support the headers. The new work is protected to some extent by being hollow, with a provision for air circulation, and has stood well for a much longer period than in the case of the old castings.

In view of the fact that a recent inspection of the water side of these boilers showed the surface of the metal to be

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absolutely free from pits, rust spots, or scale, and in perfect condition, a description of the method employed for testing water during the past two years might be of interest.

The litmus-paper test for acidity was found unsatisfactory, and at the suggestion of Mr. J. M. Cratty, chemist in the Navy Yard, Norfolk, phenolphthalein, a well known indicator in chemical analysis, was tried.

Half the amount that can be put on a ten cent piece, dissolved in 50 c.c. of alcohol, would be sufficient to last a ship for a year.

A drop of the solution in a test tube of the water to be tried will turn the whole red, if it is alkaline; but if the water is acid it will show no change in color.

The red color was generally found, but when this was not the case a small amount of soda added to the water in the boiler was sufficient to neutralize the acid.

In practice it would be advisable to have a solution of soda of such strength that a certain quantity, say one-half c.c., would have the same effect upon an ounce of water as a pound of soda would have on a thousand gallons. In this way the amount of soda required for any boiler could be readily determined.

A simple quantitative analysis for determining the chlorides in the boiler water has also been used. With a condenser ever so tight a small amount of salt will creep into the boilers after a time. When it is necessary to pump out a boiler for any purpose it is well to know if the water is good enough to use over, as feed water, or not. The hydrometer will not show this, but a small outfit with which to perform a simple chemical analysis will tell. Required are the following: A measure in ounces or c.c.; one burette, or dropper, graduated, preferably in c.c.; a standard solution of nitrate of silver; a solution of potassium chromate as an indicator; a white china dish or cup (an ordinary teacup will answer).

Have a solution of nitrate of silver of such strength that one c.c. of it will "kill" or react with one per cent. as much salt as is found in one ounce of ordinary sea water. As an illus-

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tration, take one ounce of water from a boiler, place it in the cup, add a few drops of potassium chromate, which will give it a yellow color, and drop in slowly the standard solution of nitrate of silver. The water is to be kept stirred more or less according as the silver is dropped rapidly or slowly. A red spot will appear where the silver drops, but will at once dissipate until, say, 2½ c.c. of silver solution have been added, when the whole quantity of water will turn a dark red. The result of this test shows that there is 2½ per cent. as much salt in the water tested as would be in the same quantity of sea water. With a hydrometer this water would show no concentration. In the above case the water might well be used as feed water, but if the percentage were as much as four or five, it has been the custom to pump it overboard.

The outfit required for these two simple tests is not expensive, and could be readily supplied by the chemical laboratory at the Navy Yard. Such an outfit should be added to the allowance for steam engineering of every ship.

The standard solution of nitrate of silver can be made by dissolving 28 grams of crystals in one liter of pure water. The alcohol for dissolving the phenolphthalein must be neither acid nor alkaline for delicate tests. It will generally be found slightly acid, but this can be corrected by adding a small amount of caustic soda.

RECENT PERFORMANCE.

The above article was written in June of this year, and until August 14 there was no opportunity to test the effect of the work done in cleaning the fire side of the tubes. On that date a full-power trial of four hours duration was run. The coal used was good, but it is thought to have been no better than that used during a similar run in April. The following extracts from the record of these two runs show the results obtained:

	April 15.	August 14.
Collective I.H.P. for main and auxiliary machinery	2,476	2,767.6
Coal per hour, pounds	6,435	5,830
per square foot of grate	29.2	26.5
I.H.P.	2.60	2.10

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The engines made an average of 194.7 revolutions for the four hours, working up from 140 revolutions at the time the record began. The last two hours the revolutions were 198 per minute. Much more steam could have been easily supplied by the boilers, and more could have been used by the engines, but the reducing valves in the main steam line would not permit of its passage. When this performance is compared with the trial record of the ship it will be seen that the boilers are still able to supply the demands of the engines, although they develop a great deal more power, owing to foul bottom and increased displacement.

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U. S. S. *CHARLESTON.*

DESCRIPTION AND OFFICIAL TRIAL.

By Lieutenant M. E. Reed, U. S. Navy, Member.

The *Charleston* (Protected Cruiser No. 22), built by the Newport News Shipbuilding and Drydock Company, Newport News, Virginia, is one of the three twin screw, protected cruisers authorized by Congress in an Act approved June 7, 1900, and is the first of this class to have her trials. The other two sister vessels are the *St. Louis* and the *Milwaukee*, building, respectively, at Neafie & Levy Ship and Engine Building Company's Works, Philadelphia, Pa., and at Union Iron Works, San Francisco, Cal.

The contract for the *Charleston*, signed March 30, 1901, provided for delivery of the completed vessel within thirty-six (36) months from that date; that is, on or before March 30, 1904. This time limit was extended by the Department, however, first on November 11, 1903, for a period of twelve (12) months, to March 30, 1905; second on September 1, 1904, for a period of two (2) months and fifteen (15) days, to June 9, 1905, the time from the expiration of this last extension to the final delivery of the vessel to the Government being still subject to adjustment by the Department. With these settled extensions a total allowed time before delivery accrued amounting to fifty (50) months and fifteen (15) days. The usual time penalties for failure to complete the work within the contract time are stipulated in the contract.

The total price to be paid for the vessel, exclusive of certain portions of the armor, armament, etc., which are supplied by the Government, is two million seven hundred and forty thousand dollars ($2,740,000), of which one million two hundred and forty thousand dollars ($1,240,000) was allotted for

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the building and complete installation of the machinery in readiness for service.

The guaranteed speed of the *Charleston* is 22 knots per hour maintained for four consecutive hours, on a mean draught of 22 feet 6 inches, corresponding to a displacement of about 9,700 tons. The designed power to produce this speed is about 21,000 collective indicated horsepower of propelling machinery at about 133 revolutions per minute of the main engines. Failure to reach the guaranteed speed of 22 knots under the stipulated trial conditions entailed penalties at the rate of fifty thousand dollars ($50,000) per quarter knot for speeds between 22 knots and 21½ knots, and at the rate of one hundred thousand dollars ($100,000) per quarter knot for speeds between 21½ knots and 21 knots. In case of failure to reach a speed of 21 knots it was optional with the Secretary of the Navy to reject the vessel or to accept her at a reduced price and on conditions to be agreed upon.

The keel of the *Charleston* was laid January 30, 1902, ten months after the signing of the contract; the hull was launched January 23, 1904, two years nine months and twenty-three days after the signing of the contract and one year eleven months and twenty-three days after the laying of the first keel plate.

HULL DATA.

Length between perpendiculars, feet	424
on load water line, feet	424
over all, feet and inches	426- 4½
of straight keel, feet and inches	352- 9
Beam, extreme, feet	66
at L.W.L. (22 feet 6 inches), feet	66
molded, feet and inches	65- 1
Ratio of length to beam	6.4599
Depth, molded, main deck side at M. S., feet and inches	40- 0½
upper deck side at M. S., feet and inches	47- 6½
Draught, normal, mean, feet and inches	22- 6
Displacement, mean load (empty bunkers), tons	9,148.2
normal (22 feet 6 inches), tons*	9,700
full load (24 feet 7 inches), tons	10,839
per inch at L.W.L. (22 feet 6 inches), tons	44.6

* 650 tons coal, 50 tons reserve feed water.

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Area of midship section to L.W.L., square feet	1,312.5
L.W.L. plane (22 feet 6 inches), square feet	18,730
Wetted surface, square feet	31,850
Center of gravity of L.W.L. plane aft of M. S., feet	2.45
above bottom of keel, feet	22.5
buoyancy above bottom of keel, feet and inches	12- 8¼
aft of M. S., feet	68
Transverse metacenter above center of buoyancy, feet and inches,	12- 8¼
Longitudinal metacenter above center of buoyancy, ft. and ins.	507- 6
Coefficient of fineness, block	.536
of M. S	.884
of L.W.L	.665
cylindrical	.606
Number of frames	113
watertight compartments	183

Masts and Spars.—There are two steel military masts, the foremast located at frame 31, and the mainmast at frame 80, each fitted with a lower and an upper military top, a searchlight platform, spar topmast and signal yard. On the foremast, above the signal yard, is a lookout station fitted with guard rails and having speaking-tube connections to the deck below. On the mainmast is a small gaff for ensign hoist, etc., and this mast is also fitted for wireless telegraphy.

Decks and Bridges.—The *flying bridge*, 60 feet in length, extends across the ship at frames 32–33–34, the wings being six feet in width. Around the foremast is an extension on which are located steering wheel, helm and revolution indicators, engine telegraphs, whistle and siren pulls, voice-tube stand, all necessary means of communication and instruments for the purposes of navigation. A searchlight is located on each wing of this bridge, with control stands for electric night signals and for truck lights on the port and starboard wings, respectively. Near the mast is a small ammunition winch for hoisting to the tops.

The *forward bridge* is in the form of a triangle, the forward angle being rounded off, corresponding in plan to the forward part of the upper deck. This bridge is arranged for use in bad weather, with a protected conning and steering station having a curved front with sliding windows and doorless openings at the sides. This station encloses a set of naviga-

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tional accessories similar to those on the flying bridge. On this bridge is the chart house, fitted for the use of the navigator, with chart tables, cases for charts, instruments, etc., and having tightly-shuttered sliding windows arranged for screening lights during night work. Cage mounts are provided on this bridge for two .30-caliber automatic and two .30-caliber machine guns.

The *after bridge*, immediately above and corresponding in plan to the after end of the upper deck, is fitted for signaling and for wireless telegraph operations. There is a house amidships for this latter purpose, on the top of which are also mounted two searchlights. Four automatic .30-caliber guns are mounted on this bridge.

HEIGHTS ABOVE L.W.L. (22 FEET 6 INCHES FORWARD AND AFT).

(Designed 22 feet 6 inches W.L. is 22 feet $4\frac{13}{16}$ inches above base line.)

Masts, fore and main, top of truck light, feet and inches	108– $8\frac{13}{16}$
Rangeing platform, floor ("crow's nest," forward), feet and inches	90–$10\frac{3}{16}$
Upper military top, floor, both,* feet and inches	67– $0\frac{3}{16}$
Lower military top, floor, both,* feet and inches	56– $0\frac{3}{16}$
Searchlight platforms, both, feet and inches	75– $0\frac{3}{16}$
Flying bridge at C. L., frame 34, feet and inches	41– $9\frac{3}{16}$
at side, frame 34, feet and inches	41– $2\frac{3}{16}$
above main deck, feet and inches	23– 3
Forward bridge at side, frame 34, feet and inches	34– $2\frac{3}{16}$
After bridge at side, frame 80, feet and inches	34– $3\frac{3}{16}$
Upper deck at side, M. S., frame 57, feet and inches	25– $1\frac{1}{16}$
Flat armor deck, top, feet and inches	3– 5
Main deck planking at stem, top, feet and inches	21– $3\frac{15}{16}$
stern, top, feet and inches	20– $3\frac{11}{16}$
Waist plate at stem, top, feet and inches	24–$11\frac{13}{16}$
Beading at stern, top, feet and inches	20– $9\frac{3}{16}$
Wireless telegraph station, top, feet and inches	42– $7\frac{15}{16}$
Smoke pipes, top, feet and inches	77– $7\frac{3}{16}$
Axes of 6-inch guns, gun deck, feet and inches	14– $6\frac{3}{16}$
main deck, feet and inches	21– $9\frac{3}{16}$

The *upper deck* extends from frame 28 forward to frame 87 aft, being cut away at the corners to give greater arc of fire to the 6-inch guns forward and aft, and at the sides near the cor-

* Depth of military tops, 2 feet 6 inches.

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ners to give direct ahead and astern fire, respectively, to the two pairs of main-deck 6-inch guns forward and aft. Except under the bridges, where hammock nettings are fitted, the rail of this deck is clear, and is fitted only with stanchions and life lines. A day cabin for the commanding officer is built on this deck under the forward bridge, corresponding in size to the chart house above it, and fitted with bunk, settee, lockers and toilet stand. The boats, except two life boats, are stowed on this deck when at sea, in chocks and cradles at the eight positions convenient for handling by the four boat cranes when swung to their extreme forward and extreme aft train.

Type.		Location.	Handled by —— crane in —— position.	Weight, pounds, in service, with crew.
Steam cutter,	36-foot.	S.	10-ton, S., aft.	16,961
Steam cutter,	33-foot.	P.	10-ton, P., forward.	14,436
Sailing launch,	36-foot.	P.	10-ton, P., aft.	9,954
Sailing launch,	33-foot.	S.	10-ton, S., forward.	8,471
Cutter,	30-foot.	S.	3-ton, S., forward.	5,625
Cutter,	30-foot.	P.	3-ton, P., forward.	5,625
Cutter,	26-foot.	S.	3-ton, S., forward.	4,447
Cutter,	26-foot.	P.	3-ton, P., forward.	4,447
Cutter,	26-foot.	S.	3-ton, S., aft.	4,447
Cutter,	26-foot.	P.	3-ton, P., aft.	4,447
Cutter,	26-foot.	S.	10-ton, S., forward.	4,447
Cutter,	26-foot.	P.	10-ton, P., aft.	4,447
Whaleboat,	30-foot.	S.	On sea davits.	4,802
Whaleboat,	30-foot.	P.	On sea davits.	4,802
Whaleboat (gig),	30-foot.	S.	3-ton, S., aft.	4,471
Dinghy,	20-foot.	S.	3-ton, S., aft.	2,080
Dinghy,	20-foot.	P.	3-ton, P., aft.	2,080
Dinghy,	14-foot.	S.	3-ton, S., aft.	1,325
Punt,	14-foot.	S.	3-ton, S., forward.	700 (about).
Punt,	14-foot.	P.	3-ton, P., forward.	700 (about).

The engine-room and fireroom hatch enclosures and all other trunks or companion ways end at the upper deck with suitable coamings, gratings and skylights or canopies.

In connection with the lower compartments there are on this deck: eight fireroom ventilators, in pairs, port and starboard, at frames 39, 48, 56, 65; four engine-room ventilators, in pairs, port and starboard, at frame spaces 72–73, 81–82; two galley ventilators, port and starboard, at frame space 53–54;

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and three small ventilators for air supply to small electric blowers. All ventilators for natural ventilation are fitted with rotating cowl heads.

Four smoke pipes, each connected with four main boilers, rise 100 feet from the base line of the ship and about 52 feet 6 inches from the upper deck. The outside diameter of smoke pipes is 8 feet 6 inches.

Four electrically-operated boat cranes are placed on this deck, their heels being stepped on the second (gun) deck below. The forward pair, at frame space 47–48, port and starboard, are designed for 3 tons dead-lift capacity; the after pair, at frame space 66–67, port and starboard, being designed for 10 tons dead-lift capacity. The boat-hoist duty of these cranes has been shown in boat table, and their motors will be referred to elsewhere.

Four electrically-operated deck winches for coaling and general duty are located on this deck, in pairs, port and starboard, at frame space 44–45 and 67–68.

Two electrically-operated ammunition-hoist winches for hoisting to the tops are located, one below the after side of the forward bridge and one below the forward side of the after bridge.

Twelve 3-inch R. F. guns are mounted on this deck, in pairs, port and starboard, at frames 28, 31, 45, 69, 82, 86, on regulation cage mounts; two 1-pounder R. F. guns, port and starboard, at frame 36; and two automatic .30-caliber guns, port and starboard, at frame space 78–79.

A horizontal, cylindrical fresh-water tank of 2,000 gallons capacity is located on this deck, having a dividing diaphragm fitted at its midlength, separating the tank into two equal compartments, port and starboard, the axis of the tank being athwartships. This tank is piped for gravity supply to the galley, bakery and those lavatories and pantries below which are not supplied by separate tanks, and is fitted with locked manhole plates, water glasses and overflow vents.

The *main deck* is the first of the continuous decks extending the whole length of the ship. Forward on this deck are the four deck openings of the hawse pipes for bower and sheet

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chains, two riding bitts for anchor chains, a double-headed towing bitt amidships, four chain controllers with lever throw-outs, two anchor davits for handling the Navy-type anchors to the bill boards, bow stoppers, ring bolts, etc.

In a covered pocket between the main and gun decks, and rising above the main deck to a level below the axis of the forward 6-inch gun, is located the anchor windlass, the chains passing to the wildcats through chain pipes in the forward side of this pocket.

Leadsman's platforms are located outside the rail at frame 17, port and starboard.

Two electrically-operated deck winches for anchor work, boat hoisting, coaling and general duty are located at frame 26, port and starboard.

Within the superstructure on the main deck is berthing space for the crew, hammock nettings being fitted throughout at sides where not in wake of battery, doors, etc.

Aside from the trunks for ventilation, communication and access to lower compartments there are on this deck the following:

General mess issuing room, around foremast;

Bakery, forward of second smoke pipe, amidships;

Galleys, officers' and crew's, between second and third smoke pipes;

Wash-deck gear room, port side, opposite fourth smoke pipe;

Bread room, starboard side, opposite fourth smoke pipe;

Paymaster's office, forward side engine-room hatch;

Battalion lockers, after side engine-room hatch;

Executive officer's office, after compartment, starboard side;

Navigator's office, after compartment, port side.

Aft, on the main deck, outside the superstructure, are the usual companionways, skylights and ventilating trunks; also one electrically-operated general-service winch at frame 94, amidships.

Suitable warping and towing chocks, and double-headed

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warping and towing bitts are distributed in convenient positions, forward and aft, for warping and towing.

The battery on this deck, six 6-inch guns and six 3-inch guns, is arranged as described under *Armament.*

Boat davits are fitted for two 30-foot cutters and for two 20-foot dinghies abreast of forward end of superstructure, and for two 30-foot whale boats abreast of after end of superstructure, all in pairs, port and starboard.

The *gun deck* is arranged as follows from forward:

Crew's shower baths;
Crew's water closets;
Windlass engine and windlass compartment (pocket);
Petty officers' and machinists' water closets;
Berthing spaces for crew;
Commissary's issuing room, port;
Paymaster's issuing room, port;
Laundry, starboard;
Upper casemate, armored;
Armory, forward side engine-room hatch;
Officers' lavatories, port;
Wardroom officers' rooms, four starboard, two port;
Ordnance officer's office, starboard;
Wardroom mess room, athwartships;
Captain's office, pantry, cabins (main and after), staterooms (two) and baths (two).

On this deck are eight 6-inch guns, twelve 3-inch guns and two 1-pounder R.F. guns arranged as described under *Armament.*

The *berth deck* is arranged as follows:

Lamp room, paint room (port), oil room (starboard), prisons (two), all in forward compartment;

Master-at-arms' room (port), band room (starboard), sergeant of marines' room (port), isolation ward (starboard), crew's berthing space, all in second compartment;

Crew's berthing space, chain lockers, in third compartment;

Sick bay, sick-bay bath room (port), dispensary (starboard), operating room (starboard), all in fourth compartment;

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Chief machinists' quarters and lavatory (port), chief petty officers' quarters and lavatory (starboard), crew's berthing space, all in fifth compartment;

Coal bunkers B–101, B–105, B–111, B–115 (starboard), B–102, B–106, B–110, B–114 (port), blacksmith shop (forward end of first uptakes), firemen's wash rooms (three, amidships, forward and after end of second uptakes and forward end of third uptakes), evaporator room, distillers in trunk above (between third and fourth uptakes), servants' wash room (after end of fourth uptakes), general work shop (port), refrigerating rooms (starboard), engineer's office (central, forward end of engine-room hatch), ice-machine room (starboard), ordnance, navigator's, warrant officers' storerooms (three, port), wardroom, marines', junior officers' storerooms (three, starboard), warrant officers' pantry (port), junior officers' pantry (starboard), all within the lower armored casemate, sixth compartment.

Warrant officers' quarters, mess room and five rooms (port), and one additional room (starboard), junior officers' quarters, mess room and four rooms (starboard), all in seventh compartment;

Wardroom rooms (six port and six starboard), in eighth compartment;

Captain's storerooms (two, port and starboard), blower room (right aft), in ninth compartment.

Electrically-operated ventilating blowers are located on this deck, as described under *Ventilation.*

The *protective deck* extends throughout the length of the vessel, being flat amidships and sloping downward to the sides and ends below the water line. The deck level corresponds with that of the berth deck, the spaces between the two decks at the sides and ends of the vessel being utilized for stores appropriate to the berth-deck spaces over them. The coal bunkers mentioned under the heading *berth deck* extend to the protective deck and are in communication with the lower bunkers under them by suitable doors in the protective deck.

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The *upper platforms* are directly below the protective deck and are arranged for stores, magazines, ammunition passages, steering gear, etc. From frame 7, forward, these compartments are, in order, as follows:

General stores;
Bread room;
Ordnance storeroom (port);
Commissary's storeroom (port);
Paymaster's storerooms (two, port);
6-inch magazines (two, port and starboard);
Communicating room (central);
6-inch shell rooms (two, port and starboard);
Upper dynamo room (central);
Ammunition passages (two, port and starboard through boiler spaces, then one, central, through engine space);
6-inch shell room (port);
6-inch magazine (starboard);
6-inch magazine (athwartships);
Steering-engine room;
Steering-gear room.

The *lower platforms* are next below the upper platforms and are also arranged for stores, magazines, etc. Between frames 13 and 30, forward, are:

Construction storerooms (two port and three starboard);
Equipment storeroom (port);
6-inch magazine (athwartships);
3-inch magazine (port);
3-pounder and 1-pounder ammunition room (starboard);
Small arms ammunition room (starboard);

Between frames 83½ and 99, aft, are:

Engineers' storerooms (two, port and starboard);
Small arms ammunition rooms (two, port and central);
3-pounder and 1-pounder ammunition room (port);
6-inch magazine (starboard);
Dry-provisions rooms (two port, three starboard);
Saluting-powder room (starboard);
Sail room (port).

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The *hold* is arranged as follows:

Trimming tanks (two, first and second compartments);
Store space, including fresh water tanks (third compartment);
3-inch magazine (fourth compartment);
Electrical storerooms (two, port and starboard);
Lower dynamo room (central);
Boiler rooms (four central, across center line);
Coal bunkers B–5, B–7, B–9, B–11 (starboard), B–6, B–8, B–10, B–12 (port);
Engine rooms (two, port and starboard);
Shaft alleys (two, port and starboard);
Engineers' storerooms (two, port and starboard);
3-inch magazine (central);
Water tanks (athwartships);
Trimming tanks (two, after compartments).

The *double-bottom compartments* are described hereafter under watertight compartments.

Framing.—All hull material not otherwise specified will be of mild steel of domestic manufacture.

The longitudinal framing consists of a vertical keel, five longitudinals on each side of this keel and below the protective deck, one stringer each side behind the armor about midway between protective and berth deck, and bilge keels at the fourth longitudinals (port and starboard). The *vertical keel* is 39 inches in depth except forward of frame 7 and abaft frame 102, where it tapers to its connection with the stem and stern post, and is continuous throughout. The plating of this keel is 20 pounds per square foot from frame 23 to frame 84 and 17½ pounds forward of frame 23 and aft of frame 84. The keel is not made watertight except within the space devoted to reserve fresh water where the compartments are divided, port and starboard, all other midship compartments extending across the center line so far as watertightness is concerned. Lightening holes are cut in the vertical keel forward and abaft the limits of the double bottoms. The *longitudinals* are continuous throughout the double bottoms and are carried as far to the ends of the vessel as consistent with

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good work. Within the limits of the double bottoms they are of 17½-pound plate, and beyond this limit they are of 15-pound plate. The third longitudinal only, port and starboard, is made watertight, the others being pierced with lightening holes, drainage holes being provided to allow water to flow to the lowest pockets of compartments. The *stringer* behind armor is worked with 15-pound plates between frames and with a continuous Z-bar 6 inches by 3 inches by 3 inches across the inside of the frames. It extends forward to the stem, where it forms a breast hook, and aft as far as possible. The *bilge keels* are composed of two 15-pound plates about 24 inches in width forming a sharp V section, the inner space of which is filled in with yellow pine, and the outer edges of which are finished and stiffened by 2½-inch half-round bars, one on each side of the seam. These keels extend about 200 feet amidships and are secured to the outside plating over the fourth longitudinal. The *stem* is of cast steel, in two pieces, scarfed at about the normal load line and fitted to receive the outside plating, keel plates, deck and breast-hook connections. The *sternpost* is also of cast steel, fitted to receive plating, keel plates, etc., and for carrying the rudder. Provision is made for tiller-post stuffing box, packing and gland, hardover stops for rudder, and bushed lug for rudder pintle.

The *transverse frames*, 113 in number, are spaced 3 feet 6 inches apart from the stem to frame 35 and from frame 84 to the stern. Between frames 35 and 84, the boundaries of the boiler and machinery spaces, the frames are spaced 4 feet apart. In all cases intermediate frames are worked where required for additional strength or stiffness. The framing of the superstructure above the line of the main deck is not continuous with the structure below, but is secured to the main-deck plating by bracket plates.

Plating.—In general, all *outside bottom plating*, below the line of the protective deck, is raised and sunken, with lapped butts, except where doubled. This plating is of 23 pounds per square foot for a distance of about 200 feet amidships and is reduced to 20 pounds at the ends in two reductions. The

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outside topside plating, above the line of the protective deck, is worked flush and is of 30 pounds per square foot behind armor diminishing to 20 pounds at the ends in five reductions. The *flat keel plates* are worked flush in two thicknesses, the inner plate of 23 pounds per square foot reduced at the forward end to 20 pounds, and the outer plate of 25 pounds per square foot reduced at the forward end to 22 pounds, the reduction of these plates forward being made in two steps.

Watertight Compartments.—There are 183 watertight compartments, of which 120 are below and 63 are above the protective deck.

Of those below the protective deck B–1, B–2, B–3, B–4 are boiler compartments; C–1, C–2 are engine compartments; D–2, D–3 are shaft-alley compartments; D–4, D–5, D–12, D–13 are engineer storerooms; B–73 to B–99, inclusive, are double-bottom compartments under boiler rooms (of which B–73 to B–88, inclusive, are wing compartments under coal bunkers and outboard sides of boiler rooms); B–89 to B–93, inclusive, are central compartments extending between the third longitudinals, port and starboard, and non-watertight at center line; B–94, B–95, B–96, B–97, B–98, B–99 are alternate port and starboard compartments fitted for *reserve feed-water tanks*, the entrance manholes being in the after two firerooms and of the high-coaming type); C–91 to C–99, inclusive, are double-bottom compartments under engine rooms; B–5, B–6, B–7, B–8, B–9, B–10, B–11, B–12 are lower coal bunkers abreast of boiler rooms.

Of those above the protective deck B–100 contains the hatch enclosures and passages for the forward two boiler rooms; B–109 contains the hatch enclosures and passages for the after two boiler rooms and includes the evaporator room; B–101, B–102, B–105, B–106, B–110, B–111, B–114, B–115 are upper coal bunkers over boiler rooms; C–106 contains the engine-hatch enclosure; that portion of C–100 below the berth-deck level between frames 68 and 79 at port side is assigned as additional space for engineering stores and is reached through scuttles in the machine shop and ordnance storeroom.

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COAL-BUNKER CAPACITIES.

(Measured to 6 inches below bulb of deck beams, at 43 cubic feet per ton.)

Bunkers in hold.	Cubic feet.	Tons.	
B– 5	3,417	79.5	
B– 6	3,417	79.5	
B– 7	4,795	111.5	
B– 8	4,795	111.5	
B– 9	5,296	123.2	
B–10	5,296	123.2	
B–11	5,427	126.2	
B–12	5,427	126.2	
			880.8
Bunkers above protective deck.			
B–101	4,426	102.9	
B–102	4,426	102.9	
B–105	4,306	100.1	
B–106	4,306	100.1	
B–110	4,399	102.3	
B–111	4,399	102.3	
B–114	4,285	99.7	
B–115	4,285	99.7	
			810
Total,	72,702		1,690.8

DOUBLE-BOTTOM AND RESERVE-FEED WATER-TANK CAPACITIES.

Compartment.	Gallons.	Salt water, tons.	Fresh water, tons.
A–97	7,622.12	29.11	28.29
A–98	18,535.34	70.79	68.8
A–99	11,738.79	53.86	52.46
B–89	14,373.8	65.95	64.24
B–90	11,751.86	53.92	52.52
B–91	11,950.2	54.83	53.4
B–92	12,122.38	55.62	54.17
B–93	12,205.2	56	54.54

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Compartment.	Gallons.	Salt water, tons.	Fresh water, tons.
B–94 (Reserve feed)	6,115.68	28.06	27.33
B–95 (Reserve feed)	6,115.68	28.06	27.33
B–96 (Reserve feed)	6,085.16	27.92	27.19
B–97 (Reserve feed)	6,085.16	27.92	27.19
B–98 (Reserve feed)	7,519.28	34.5	33.6
B–99 (Reserve feed)	7,519.28	34.5	33.6
	39,440.24	180.96	176.24
C–97	14,347.65	65.83	64.12
C–98	13,506.36	61.97	60.36
C–99	12,811.1	58.78	57.25
D–98	9,372.44	35.8	34.8
D–99	6,359.96	25.06	24.36
Total,	196,137.54	868.48	845.56

TRIMMING-TANK CAPACITIES.

Compartment.	Gallons.	Salt water, tons.	Fresh water, tons.
A– 1, forward,	9,910.19	45.47	44.29
A– 2, forward,	14,402.14	66.08	64.36
D– 9, aft,	10,389.68	47.67	46.43
D–10, aft,	12,011.25	55.11	53.68
Total,	46,713.26	214.33	208.76

FRESH-WATER TANK CAPACITIES.

Compartment	Tank.	Gallons.	Tons.
A–3	13–15, Port,	816	3.04
A–3	13–15, Starboard,	816	3.04
A–3	16–18, Port,	1,484	5.53
A–3	16–18, Starboard,	1,484	5.53
A–3	19–20, Port,	1,535	5.73
A–3	19–20, Starboard,	1,535	5.73
D–8	Port,	1,182	4.41
D–8	Starboard,	1,182	4.41
Upper deck,	Port gravity,	1,000	3.71
Upper deck,	Starboard gravity,	1,000	3.71
	Total,	12,034	44.84

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Coffer Dams.—A continuous coffer-dam belt about 30 inches wide extends the entire length of the ship at sides, from frame 7 forward to frame 109 aft, occupying the space on the slope of the protective deck outboard and rising to the level of the berth deck. The compartments comprised in this belt are: A–101, –104, –105, –108, –109, –112, –113, –116, –117; B–103, –104, –107, –108, –112, –113, –116, –117; C–102, –103, –104, –105; D–102, –103, –106, –107, –109, –110.

After the preliminary trials of the vessel these coffer dams are to be packed with suitable fire-proofed, corn-pith cellulose having the compressed surfaces laid in the horizontal plane.

Armor and Protection.—On the 20-pound plating of the *protective deck* are worked nickel-steel plates 1½ inches thick on the flats and 2½ inches thick on the slopes, giving a total thickness of 2 inches on the flats and 3 inches on the slopes. All hatches and openings through the protective deck are fitted with shutters of the same thickness as the deck, or with armor bars of approved strength.

The *main-belt side armor* extends between frames 35 and 84, a distance of 196 feet, abreast of the boiler and engine compartments. This belt is 4 inches in thickness and 7 feet 6 inches in height, of which 3 feet 5 inches is above the 22 feet 6 inches load water line. The upper edge of this belt reaches to the berth-deck level, the athwartships triangular spaces between the protective deck and berth deck at the ends of the side armor being filled in with 3-inch armor.

The *lower casemate*, extending from berth deck to gun-deck level, corresponds in thickness and area to the main side-armor belt, the side armor being 4 inches thick, and the athwartships armor at ends being 3 inches thick.

The *upper casemate* extends from gun deck to main-deck level, and, like the lower casemate, is composed of 4-inch side armor and 3-inch athwartship armor at ends. The length of this casemate is only 128 feet, however, as compared with the 196 feet of the lower casemate and main belt, extending only between frames 41 and 73 with an extension of the side armor at the corners over the next frame space forward and aft,

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making the sides 136 feet in length. The after athwartships armor of this casemate coincides with the forward end of engine-room hatch at that level, while the forward athwartships armor coincides with the after side of the uptake enclosure for the forward group of boilers at that level. The upper casemate encloses the eight 6-inch gun positions on the gun deck, the gun positions being separated by 2-inch nickel-steel splinter bulkheads extending inboard about 15 feet from the side.

An armored *conning tower* is located at frame 28. It is elliptical in shape with its long axis athwartships, and has its entrance at the rear protected by a concentric shield plate which gives an 18-inch access passage, having the same curve as the tower. The height of the conning tower is 7 feet 6 inches and its axes are 6 feet and 8 feet, respectively. The thickness of both tower and shield plate is 5 inches. The top of the conning tower is covered by a 3-inch plate and of the access passage by a 3-inch sloped plate, the bottom plate, 2 inches thick, being shaped to include the access passage and to support the shield plate.

An *armored tube*, 30 inches inside diameter and 3 inches thick, is fitted from the floor of the conning tower to the protective deck. It provides for the leads of voice pipes, wiring, etc., for the various conning-tower devices and also affords access to the conning tower from the communicating room, ladder rungs being fitted inside the tube for this purpose.

Four *sponsons* of 4-inch armor give local protection to the four 6-inch gun positions at the corners of the superstructure on the main deck. The armor extends from the top of the upper casemate to the upper-deck level and is curved inward around the gun positions to afford protection in a fore-and-aft direction.

The twelve 3-inch gun positions on the gun deck outside of the upper casemate are protected by 3-inch nickel-steel *port plates* about 18 feet 6 inches long, extending over the space between the gun and upper decks. The forward and after 6-inch gun positions, on main deck outside of superstructure,

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and the remainder of the secondary-battery positions not previously mentioned, are not provided with armor protection.

Armament.—The *main battery* consists of fourteen 6-inch rapid-fire rifles, 50 calibers in length. Two of these are mounted, one forward and one aft, on the main deck outside the superstructure, on the center line, at frames 23 and 91, respectively, each having a total arc of train of 290 degrees. Four are mounted on the main deck in armored sponsons within the corners of the superstructure, with an arc of train of 90 degrees forward and 35 degrees abaft the beam for the forward pair and 35 degrees forward and 90 degrees abaft the beam for the after pair. The remaining eight are mounted on the gun deck within the upper casemate, in box battery, four on each broadside, with separating splinter bulkheads, as before mentioned; the forward and after pairs of this arrangement are mounted in sponsons at the corners of the upper casemate directly below the corresponding pairs on the main deck, and have a total arc of train of 120 degrees, 85 forward and 35 abaft the beam for the forward pair, and 35 forward and 85 abaft the beam for the after pair; the total arc of train of the second and third pairs of this arrangement is 112 degrees, 77 forward and 35 abaft the beam for the second pair, and 35 forward and 77 abaft the beam for the third pair. The mean height of the axis of main-battery guns above the 22 feet 6 inches L.W.L. is about 21 feet 9 inches for the main-deck battery and 14 feet 6 inches for the gun-deck battery.

The *secondary battery* consists of eighteen 3-inch, rapid-fire guns 50 calibers in length; twelve 3-pounder semi-automatic guns; four 1-pounder heavy automatic guns; eight 1-pounder heavy rapid-fire guns; two 3-inch field guns; two machine guns, .30-caliber; and eight automatic guns, .30-caliber. Twelve 3-inch guns are mounted on the gun deck, outboard, outside the upper casemate, in frame spaces 13–14, 23–24, 31–32, forward, and 82–83, 91–92, 101–102, aft, in pairs, port and starboard. These twelve guns have 3-inch armor port-plates, as before mentioned. The remaining six 3-inch guns are mounted outboard on the main deck within the superstructure, at frame spaces 52–53, (on) 57, 61–62, in pairs, port

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and starboard. These guns have no armor-port plates. The 3-pounder guns are equally divided on each side of the vessel and so disposed in commanding positions as to give the largest arc of fire without interference with each other or with the other guns of the battery. Under the original disposition of the battery two 1-pounder heavy automatic guns were to be mounted in each lower military top, fore and main; two 1-pounder heavy R.F. guns in each upper military top, fore and main; two of the remaining four 1-pounder heavy R.F. guns aft on the gun deck; and two 1-pounder heavy R.F. guns forward on the upper deck. A new disposition is now to be given to the guns formerly assigned to locations in the military tops in order to devote these tops to range-finding purposes. The field, automatic and machine guns have field-carriage mounts and cage-stand or rail-socket positions.

BATTERY ARRANGEMENT AND TRAIN.

MAIN BATTERY.

Fourteen 6-inch, 50-caliber R. F. guns.

Number.	Location.	Train, forward and abaft beam.
Nos. 1 and 2.	Gun deck, frame 44, S. and P.,	85 ° fwd., 35 ° aft.
Nos. 3 and 4.	Gun deck, frame 52½, S. and P.,	77 ° fwd., 35 ° aft.
Nos. 5 and 6.	Gun deck, frame 61½, S. and P.,	35 ° fwd., 77 ° aft.
Nos. 7 and 8.	Gun deck, frame 70, S. and P.,	35 ° fwd., 85 ° aft.
No. 9.	Main deck, frame 23, central,	ahead to 55° aft, P. and S.
Nos. 10 and 11.	Main deck, frame 44, P. and S.,	90 ° fwd., 35 ° aft.
Nos. 12 and 13.	Main deck, frame 70, P. and S.,	35 ° fwd , 90 ° aft.
No. 14.	Main deck, frame 91, central,	astern to 55° fwd., P. and S.

SECONDARY BATTERY.

Eighteen 3-inch, 50-caliber, R. F. guns.

Nos. 1 and 2.	Gun deck, frame 13½, S. and P.,	90 ° fwd., 24 ° aft.
Nos. 3 and 4.	Gun deck, frame 23½, S. and P.,	77 ° fwd., 43 ° aft.
Nos. 5 and 6.	Gun deck, frame 31½, S. and P.,	73 ° fwd , 41 ° aft. 76 ° fwd., 44 ° aft.
Nos. 7 and 8.	Gun deck, frame 82½, S. and P.,	42½° fwd., 75 ° aft. 43½° fwd., 73½° aft.
Nos. 9 and 10.	Gun deck, frame 91½, S. and P.,	43 ° fwd., 77 ° aft. 43½° fwd., 76 ° aft.
Nos. 11 and 12.	Gun deck, frame 101½, S. and P.,	24 ° fwd., 90 ° aft.
Nos. 13 and 14.	Main deck, frame 52½, S. and P.,	60 ° fwd., 60 ° aft.
Nos. 15 and 16.	Main deck, frame 57, S. and P.,	60 ° fwd., 60 ° aft.
Nos. 17 and 18.	Main deck, frame 61½, S. and P.,	60 ° fwd., 60 ° aft.

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Twelve 3-pounder, semi-automatic guns.

Nos. 1 and 2.	Upper deck, frame 28–29, S. and P.	
Nos. 3 and 4.	Upper deck, frame 30–31, S. and P.	
Nos. 5 and 6.	Upper deck, frame 45, S. and P.	
Nos. 7 and 8.	Upper deck, frame 69, S. and P.	
Nos. 9 and 10.	Upper deck, frame 81–82, S. and P.	
Nos. 11 and 12.	Upper deck, frame 86, S. and P.	

Twelve 1-pounder, heavy, R. F. guns.

Nos. 1 and 2.	Gun deck, frame 110½, S. and P.,	after cabin.
Nos. 3 and 4.	Upper deck, frame 36, S. and P.	
*Nos. 5 and 6.	Forward lower military top.	
*Nos. 7 and 8.	After lower military top.	
*Nos. 9 and 10.	Forward upper military top.	
*Nos. 11 and 12.	After upper military top.	

Eight .30-caliber Colt automatic guns.

Nos. 1 and 2.	Upper deck, frame 78–79, S. and P.
Nos. 3 and 4.	Forward bridge, frame 29, S. and P.
Nos. 5 and 6.	After bridge, frame 82, S. and P.
Nos. 7 and 8.	After bridge, frame 85, S. and P.

Two machine guns (Gatling), .30-caliber.

Nos. 1 and 2.	Forward bridge, frame 31, S. and P.

Two 3-inch field pieces.

Nos. 1 and 2.	Stowed on upper deck, frame 41–42, S. and P.

Ammunition Hoists.—There are twenty-seven electrically-operated ammunition hoists fitted to furnish an adequate supply of ammunition to the battery. Frictional slip gears are provided between the electric motors and the hoists to prevent damage being done to either in case of sudden stoppage of the hoists. Stops are fitted to prevent ammunition falling back in the hoists, these stops being hinged to turn up out of the way when striking ammunition below. The motors are reversible and operate at about 1,000 R.P.M., all necessary rheostats and switches being provided. Gear for operating the hoists by hand is also fitted. Ammunition for the guns in the fighting tops was to be whipped to the tops from the upper deck by electric whip hoists with specially-fitted swinging davits.

* Guns marked (*) are subject to re-location on deck.

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Ammunition is to be delivered at the following rates:

For each 6-inch gun	6 complete rounds every minute.
For each 3-inch gun	15 complete rounds every minute.
For each 3-pdr. gun	30 complete rounds every minute.
For each 1-pdr. R.F. gun	30 complete rounds every minute.
For each 1-pdr. aut. gun	100 complete rounds every minute.

Ammunition hoists, 6-inch, number and location:

No. 1 6-inch gun supplied by hoist No. 11, gun deck, frame 47, S.
No. 2 6-inch gun supplied by hoist No. 10, gun deck, frame 47, P.
No. 3 6-inch gun supplied by hoist No. 13, gun deck, frame 57, S.
No. 4 6-inch gun supplied by hoist No. 12, gun deck, frame 57, P.
No. 5 6-inch gun supplied by hoist No. 15, gun deck, frame 58, S.
No. 6 6-inch gun supplied by hoist No. 14, gun deck, frame 58, P.
No. 7 6-inch gun supplied by hoist No. 19, gun deck, frame 67, S.
No. 8 6-inch gun supplied by hoist No. 18, gun deck, frame 67, P.
No. 9 6-inch gun supplied by hoist No. 4, main deck, frame 27, P.
No. 10 6-inch gun supplied by hoist No. 8, main deck, frame 41, P.
No. 11 6-inch gun supplied by hoist No. 9, main deck, frame 41, S.
No. 12 6-inch gun supplied by hoist No. 16, main deck, frame 66, P.
No. 13 6-inch gun supplied by hoist No. 17, main deck, frame 66, S.
No. 14 6-inch gun supplied by hoist No. 25, main deck, frame 88 CL.

No. 1 3-inch gun
No. 2 3-inch gun
No. 3 3-inch gun
No. 4 3-inch gun
No. 5 3-inch gun
No. 6 3-inch gun
} supplied by hoist No. 1, gun deck, frame 23, S.
supplied by hoist No. 2, gun deck, frame 23, P.
supplied by hoist No. 3, gun deck, frame 26, S.
(these hoists near center line.)

No. 7 3-inch gun
No. 8 3-inch gun
No. 9 3-inch gun
No. 10 3-inch gun
No. 11 3-inch gun
No. 12 3-inch gun
} supplied by hoist No. 24, gun deck, frame 87, C.
supplied by hoist No. 26, gun deck, frame 94, P.
supplied by hoist No. 27, gun deck, frame 94, S.
(these hoists on and near center line.)

No. 13 3-inch gun
No. 14 3-inch gun
No. 15 3-inch gun
No. 16 3-inch gun
No. 17 3-inch gun
No. 18 3-inch gun
} supplied by hoist No. 5, main deck, frame 29, P.
supplied by hoist No. 22, main deck, frame 84, P.
supplied by hoist No. 23, main deck, frame 84, S.
(these hoists near center line.)

All 3-pdr. and 1-pdr. guns forward are supplied by hoists Nos. 6 and 7 on upper deck at frame 34, P. and S., those aft being supplied by hoists Nos. 20 and 21, at frame 84, P. and S.

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Ground Tackle.—The following anchor equipment is supplied:

Two bower anchors, Baldt type, weight (two) 23,875 pounds.
One sheet anchor, Navy type, weight, 13,076 pounds.
One stern anchor, Navy type, weight, 3,487 pounds.
One stream anchor, Navy type, weight, 2,438 pounds.
Two kedge anchors, Navy type, weight, 815 and 404 pounds.

There are 360 fathoms of Navy type, 2½-inch cable supplied, two bower chains, 120 fathoms each, and one sheet chain, 120 fathoms; the total weight of these cables being 132,000 pounds.

PROPELLING MACHINERY.

There are two four-cylinder, vertical, inverted, direct-acting, triple-expansion propelling engines placed abreast, each in its own watertight compartment, separated by a center-line bulkhead, through which is a communicating watertight door. The engines are right-and-left and are arranged for inboard-turning propellers when going ahead. The designed power of these main engines is 21,000 I.H.P. at about 133 revolutions per minute, with a steam pressure of 250 pounds at the high-pressure valve chest.

The main-engine *cylinders*, four for each engine, consist of casings of cast iron, which include the valve chests, steam ports and passages, lower heads, brackets, tie-rod and support facings, and are machined for all necessary fittings. All cylinders and valve chests are fitted with *working liners* of hard, close-grained cast iron. The cylinder liners have bearings at both ends and at about mid-length, and are secured to the cylinder casings by countersunk screws through the bottom flanges, the casing being machined to take this lower seating of the liners. There is a composition gland at the top end of the liners fitted for packing between the liner and the casing, and which allows for expansion while making a tight joint at the end. The space between the liner and the casing forms a steam-jacket space for the barrel of the cylinder. These liners are bored to accurate dimensions after being fitted and while in their natural vertical position, and are counter-bored to give the piston-packing rings a short override at the ends of the working bore. The valve-chest liners have bearings at the ends, the liner for each valve chest being in two parts, and are

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forced into place and secured by screws tapped half into the liners and half into the casings. The *cylinder covers* are of cast iron, strongly ribbed, fitted with central manholes, manhole plates and lagging covers, the I.P. and L.P. covers being fitted to be steam jacketed. The *valve-chest covers* are of cast iron, strongly ribbed, the upper covers containing the balance piston cylinders and the lower covers being bored and faced to receive the valve-stem crosshead guides and stuffing boxes. The valve chests are arranged to draw the H.P. and I.P. valves from the top for examination and the L.P. valves from the bottom.

The *order of cylinders*, beginning forward, is as follows: Forward low pressure, high pressure, intermediate pressure, after low pressure. The crank shafts are forged with the forward low-pressure crank and the high-pressure crank (which form the forward section of the complete crank shaft) opposite each other, and with the intermediate-pressure crank and the after low-pressure crank (which form the after section of the complete crank shaft) opposite each other, the first-mentioned pair being set at right angles to the last-mentioned pair. The *crank sequence* when the engines are going ahead is then: high-pressure, intermediate-pressure, forward low-pressure, after low-pressure, with a uniform angularity of 90 degrees apart.

To provide for the *framing and bracing* of the main engines the four cylinders of each engine are supported on twelve forged nickel-steel cylindrical columns, having swells and facings for the athwartship and longitudinal braces and tie rods, and flanges at top and bottom for securing to the cylinders and bed plates. Athwartships, between each pair of front and back columns, is fitted an X-brace, with a horizontal tie rod through the center of the X, on which are lateral distance pieces. Longitudinal stiffness is provided by diagonal tie rods on the outside of columns, front and back, one from top of first to foot of third column, bolted at center boss of second, and a similar one for the fourth, fifth and sixth columns. On the inside of front columns at center is secured a horizontal fore-and-aft tie rod, having an

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adjusting screw and sleeve at its midlength. On the inside of back columns at center is fitted a horizontal I-beam stiffener, to which the lower ends of the crosshead guides are secured. Additional stiffness is given to the engines by athwartship ties between pads on the cylinder bases of the two engines and by fore-and-aft ties secured to pads in the hull structure forward and abaft the engines. This arrangement gives a very stiff and rigid engine, as was shown by the severe test during the official trials.

The engine *bed plates*, consisting of two longitudinal and six cross girders of I section, are of cast steel, three sections for each engine, bolted together at flanges, and each section carrying three main bearings. The bed plates rest on and are secured to the keelson plates provided in the hull structure. Proper facings and finishings are provided for flanges of engine columns, seatings for bearing brasses and caps, and for turning engine.

All the *main steam valves* are of the single-ported piston type, with single cast-iron packing rings and steel followers. The H.P. valves are of cast iron; the I.P. and L.P. valves are built up with two heads of cast iron with a steel-tube distance piece between. The H.P. cylinder has one valve and the I.P. and each L.P. cylinder two each. There is a steam piston at the top of each valve stem, working in a short cylinder in the valve-chest cover, which is so designed as to relieve a part of the weight of the valve gear. The H.P. and I.P. balance pistons are 7 inches in diameter and the L.P. 9 inches in diameter. Steam is taken on the inside of the H.P. valves and on the outside of the I.P. and L.P. valves.

The *valve gear* is of the Stephenson type with double-bar forged-steel links, the H.P. valve stem of each engine being connected directly to its link block while the pairs of I.P. and L.P. valve stems for their respective cylinders are connected by a crosshead.

The maneuvering of the engines is controlled by a steam *reversing engine* for each, of the floating-lever type, the piston rod of which is connected to the reverse shaft carried on the

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front engine columns, and also carries a control piston in an oil cylinder giving steadiness of motion and allowing for hand operation by means of an oil pump. Independent *cut-off blocks* are fitted, one for each link, in the arm of the reverse shaft, giving a range of independently adjustable cut-off from .5 to about .73 of the stroke.

All main-engine cylinders are *steam jacketed* around the working liners, and the I.P. and L.P. cylinders are also jacketed at the ends. Live steam from the main steam pipe on the boiler side of the engine stop valves is led through the H.P. jackets and thence through the I.P. and L.P. jackets, with suitable reducing and safety valves.

Each engine is fitted with a balanced *throttle valve* having a clear opening equal to 13 inches in diameter, the balance being accomplished by a piston working in a cylinder cast in the valve casing.

The *pistons* are conical in shape, the H.P. being of cast iron and the I.P. and L.P. of cast steel. The H.P. and I.P. pistons are each fitted with one hard cast-iron packing ring 2½ inches wide and 1 inch thick, arranged for limited expansion and set out by seven U-shaped springs in the joint. There are also eighteen double-ended, flat-looped springs for setting out each H.P. packing ring, thirty-three for each I.P. ring and thirty-nine for each double L.P. ring. The L.P. pistons are each fitted with two rings 1¼ inches by 1 inch, each separated by diagonal cuts into eight sections. Composition tongue pieces, set in flush, cover these cuts between the sections and the sections of one ring break joint with those of the other, dowels preventing independent rotation of one ring on the other.

The *follower rings* of all pistons are of cast steel, secured to the piston body on square-shouldered studs by nuts and split pins. All pistons have 14½-inch bosses with outside threads for lifting and handling, and are taper-bored to receive the piston-rod end, setting up against a filleted collar on the rod, and secured from turning by a dowel in the rod. The piston rod is threaded beyond the face of the piston boss and a cylindrical nut, slotted for a spanner, sets up against the boss face

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and is locked by a ring with tongues corresponding to the slots in the nut which slips over the nut and is bolted to the boss face.

The *piston rods* are hollow, of nickel-steel, oil-tempered and annealed, accurately ground and polished to 8 inches outside diameter, the axial hole being 4 inches in diameter for the H.P. and I.P. rods and 5 inches for the L.P. rods, except at the crosshead ends where it tapers to 1½ inches for the H.P. and I.P. and 1¾ inches for the L.P. rods respectively.

The *crossheads* are of forged steel, taper-bored to receive the piston rod, and machined for the crosshead slipper of cast steel which is bolted to the flanged lug of each. The piston rod is threaded at the end beyond the face of the crosshead, and a cylindrical nut, slotted for a spanner, sets up against the bottom face of the crosshead. The crosshead pins extend on each side of the crosshead, being forged in one piece with it, are taper-bored from ends to center of crosshead, and have a ⅜-inch flat on the sides.

The *crosshead guides* are of cast iron, hollow for the circulation of water, and are secured by their bottom ends to the strongbacks on the back engine columns and at their tops to lugs on the cylinder casings. They are machined with side flanges to which the L-shaped backing guides are bolted.

The *connecting rods* are of nickel-steel, oil-tempered and annealed, having a 4-inch axial hole for the H.P. and I.P. rods and a 5-inch hole for the L.P. rods. The upper end is forked to span the crosshead and the ends of the fork are machined to take the crosshead brasses which are bolted to them with two bolts each. The lower end is machined to take the crank-pin brasses, which are bolted to it with two bolts, the bottom brass being backed by a machined, forged, flat-steel cap.

The *eccentrics* are made in two parts, the thin section of forged steel and the thick section of cast iron. These are accurately fitted together and secured by two 2-inch bolts. After being fitted to their seating they are turned on the outside to a uniform eccentricity of 5 inches. The backing eccen-

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tric of each pair is keyed to the shaft and the go-ahead eccentric is bolted to the backing eccentric by two through bolts in elongated holes, the holes being filled in solid after setting the eccentrics to prevent displacement.

The *eccentric straps* are of composition, lined with white metal, accurately fitted to the eccentrics.

The *eccentric rods* are of nickel-steel, having T heads at the lower ends for securing to the eccentric straps and forked-top ends to span the main links at the link pins, adjustable brasses being fitted for these pins.

The *main links* are nickel-steel curved bars, having the pins for suspension rods and eccentric rods forged on. Each pair of bars is securely bolted together at the ends, and the ahead half of the link is divided into ten equal divisions, from 0, the mid-position, to 10, full gear.

The *main link blocks* are of nickel-steel forgings, fitted with jaws at the ends to span the bars of the links, these jaws being provided with adjustable composition gibs conforming to the curve of the links.

All *shafting* is hollow and is of forged nickel-steel. The crank shafts have two sections, and the thrust, line, stern-tube and propeller shafts have one section each for each main engine. A worm wheel for turning the main engine is fitted over the coupling flanges between the two sections of crank shaft, one for each engine. There are six *main bearings* for each complete crank shaft, two *steady bearings* for each thrust shaft, one *spring bearing* for each line shaft, two *stern-tube bearings* for each stern-tube shaft, and one *strut bearing* for each propeller shaft. Each shaft is made watertight at the forward end of its stern tube by a stuffing box and gland having a packing space 7 inches deep by 1 inch wide, arranged to prevent rotation of the packing with the shaft, and fitted with a water-circulation pipe from behind the packing space to the engine-room bilge. A stuffing box and gland of less depth is fitted around the shaft at the bulkhead between the engine room and shaft alley.

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The *thrust bearing* is of the horseshoe type, the pedestal being of cast iron bolted down, through oblong holes, on a cast-iron sole plate which is fitted with raised lugs at the ends which are machined for fore-and-aft adjusting wedges fitted in the space between the ends of the pedestal and these lugs. The sole plate is in turn riveted solidly to the built-up foundation from the hull structure. The sides and ends of the pedestal form a trough for oil in which the lower part of the thrust collars turn, the trough being fitted with a closed water-circulating coil for cooling the oil. The boxes for the steady bearings which carry the weight of the thrust shafts are cast in one with the pedestal as are also the lugs supporting the side adjusting rods, and a stuffing box is worked around the shaft at each end of the pedestal to prevent the escape of oil. There are twelve horseshoes of cast steel, made hollow for the circulation of water through each separately, and lined with white metal on both sides, except the forward one, which is for ahead thrust only and bears only on the forward side of the forward collar. Provision is made for independent fore-and-aft adjustment of the individual horseshoes by 4-inch, steel, screwed side rods and wrought-iron clamp nuts.

The *inboard coupling* of each stern-tube shaft consists of a forged-steel collar secured to a raised seating on the end of the shaft by four steel keys; behind this solid collar is another steel collar in halves and fitting snugly around the shaft behind the shoulder of the collar seating, both collars being through bolted to the after flange of the line shaft by eight 3¼-inch fitted bolts.

The *outboard coupling* between stern tube and propeller shafts consists of a forged-steel solid sleeve, taper bored from ends to center, secured from turning on the tapered shaft ends by four fitted keys for each shaft (two for each shaft end). The shaft ends are forced into the taper bores with the keys in their seats, and tapered drawing cotters are driven through slots in sleeve and shaft ends to prevent withdrawal, the grooves around cotter slots being filled with solder and screw stop pins being let in to secure the cotters.

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ENGINE DATA.

Cylinders, number for each engine	4
H.P., diameter, inches	36
I.P., diameter, inches	59.5
L.P. (2), diameter, inches	69
Stroke of all pistons, inches	45
Valves, H.P. (1 each H.P. cylinder), diameter, inches	22
I.P. (2 each I.P. cylinder), diameter, inches	25
L.P. (2 each L.P. cylinder), diameter, inches	$27\frac{1}{8}$
Valve stems, diameter, inches	$3\frac{1}{4}$
through valve, inches	$2\frac{1}{4}$
above valve, inches	$1\frac{3}{4}$
at balance piston, inches	$1\frac{1}{2}$
Main steam pipe, diameter at throttle, inches	13
area of cross section, square inches	132.73
Volume swept by H.P. piston, mean, per stroke, cubic feet	26.073
I.P. piston, mean, per stroke, cubic feet	72.409
L.P. pistons (2), mean, per stroke, cubic feet	194.754
Net area of I.P. to H.P. piston, ratio	2.7316 to 1
L.P. pistons (2) to H.P. piston, ratio	7.3373 to 1
Piston rods, diameter, inches	8
of axial hole, H.P. and I.P., inches	4
L.P., inches	5
length, feet and inches	9- $2\frac{1}{8}$
Cylinder walls, thickness, H.P., I.P., L.P., inches,	$2\frac{1}{8}$, $1\frac{3}{4}$, $1\frac{1}{2}$
liners, thickness, H.P., I.P., L.P., inches	$1\frac{3}{8}$, $1\frac{7}{16}$, $1\frac{3}{8}$
Valve-chest liners, thickness, H.P. and I.P., inches	$1\frac{3}{8}$
L.P., inches	$1\frac{5}{16}$
Jacket space, inch	$\frac{3}{4}$
Cylinder relief valves (2 each cylinder), diameter, inches	4
Valve-chest relief valves (1 each I.P. and L.P. valve), diameter, inches	4
Connecting rod, length center to center, inches	90
diameter, upper end, inches	$8\frac{1}{8}$
lower end, inches	$9\frac{3}{8}$
axial hole, H.P. and I.P., inches	4
L.P., inches	5
Connecting-rod crosshead bolts (4), diameter, inches	$3\frac{1}{4}$
axial hole, diameter, inches	$1\frac{1}{2}$
crank-pin bolts (2), diameter, inches	$4\frac{1}{2}$
axial hole, diameter, inches	$1\frac{7}{8}$
Crosshead slipper, surface, ahead, square inches	630
backing, square inches	472.5
pin, diameter, inches	$10\frac{1}{4}$
of axial hole, tapered ends to center, ins.	6 to 2
length over all, inches	$35\frac{5}{8}$
each side crosshead, inches	$10\frac{7}{8}$
Crank shaft, number of sections each engine	2

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Crank shaft, diameter, inches	18
axial hole, inches	11½
coupling discs, inches	32
thickness coupling discs, inches	3¾
coupling bolts (8 each flange), tapered, diameter, ins.	3 9/16
journals, diameter, inches	18
length, inches	(4) 21½, (1) 31½, (1) 26½
length forward section, F.L.P. and H.P., ft. and ins.	15–11¼
after section, I.P. and A.L.P., feet and ins.	17– 8½
distance between journal centers, ft. and ins.,	5–6, 5–6, 5–11, 6–1⅝
pin, diameter, inches	18½
length, inches	22
projected area, square inches	407
axial hole, diameter, inches	12
webs, width, inches	20¾
thickness, inches	11
Thrust shaft, diameter, inches	17¼
axial hole, inches	11½
length, feet	15
steady bearings, inches	18
collars, number on each shaft	12
diameter, inches	27
thickness, inches	2
space between, inches	4
horse shoes, number, each engine	12
total ahead surface, each engine, square inches	2,568
backing surface, each engine, sq. inches	2,354
steady bearings (2 each), length, inches	18
diameter, inches	17¼
Line shaft, diameter, inches	17
axial hole, inches	11½
coupling disc, forward, inches	32
after, inches	33½
bolts (8), 1 flange, inches	3½
length, feet and inches	21– 3
Stern-tube shaft, diameter, inches	18
axial hole, inches	11
length, feet	36
Propeller shaft, diameter in strut bearing, inches	18
axial hole, inches	11
length, feet and inches	34– 6
Inboard collar coupling, diameter, inches	33½
length, inches	17½
of collar, inches	14
keys (4 each shaft), inches	14 × 2½ × 1¾
Outboard coupling, diameter, inches	23½
taper to center, 1 inch per foot, inches	18 to 15½
keys (4 each shaft), inches	32½ × 2½ × 1½

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Outboard coupling, through-gibs (2 each shaft), inches,	$23\frac{1}{2} \times 2\frac{1}{2} \times 7\frac{1}{2}$ to $6\frac{1}{2}$
length over all, feet and inches	5-$10\frac{1}{2}$
Stern-bracket bearing, diameter, inches	$20\frac{1}{8}$
length, feet	6
Stern-tube bearing, after, diameter, inches	$20\frac{1}{16}$
length, feet	6
forward, diameter, inches	20
length, feet	6
Reversing engine, diameter of steam cylinder, inches	16
oil cylinder, inches	8
stroke, inches	22
Turning engine, diameter of steam cylinders (2), inches	7
stroke, inches	5

Main Condensers.—There is one main condenser for each engine, located abreast of engine, outboard. The shell is of steel, butted with double butt straps and stiffened with T bars circumferentially, and fitted with riveted angle-iron rings at the ends to form flanges for securing the water chests. The water chests are of composition and are partially lined with rolled-zinc plates. At the inlet and discharge end a division plate is placed horizontally across the water chest to direct the flow of water through the tubes, and in this plate is fitted a valve, worked by a lever on the outside of the condenser, to allow the circulating water to pass directly overboard when this valve is opened. Steel baffle plates, fitted directly under the inlets, deflect the entering steam and distribute it over all the tubes. The tubes are of seamless-drawn Admiralty metal, not tinned, packed with brass screw glands and cotton tape, and supported by two plates, spaced 4 feet 7 inches from the tube plates, through which the tubes are passed, the tubes resting in metal thimbles of the same composition as the tubes. Two main inlet nozzles are riveted to the top of the condenser, one at each end, on the forward one being arranged seatings for the various accessory inlets. At the bottom of the condenser is a 12-inch suction opening to the main air pump and a 12-inch manhole, in the cover of which is a boss for a boiling-out connection. In the bottom of each tube plate is a hand hole to facilitate examination, and the water-chest covers are fitted with circular manholes to allow examination of tube

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CYLINDER CLEARANCE.

Engine.	Cylinder.		Volume of cylinder, cubic inches.	Clearance, in gallons of water.	Clearance, in cubic inches.	Per cent. of cylinder volume.	Linear clearance.
Starboard, . .	H.P.	Top.	45,805	55	12,705	27.7	$\frac{15}{32}$
		Bottom.	43,543	59.25	13,687	31.4	$\frac{23}{32}$
	I.P.	Top.	125,123	123	28,413	22.7	$\frac{9}{16}$
		Bottom.	122,861	129.5	29,914	24.3	$\frac{9}{16}$
	F.L.P.	Top.	168,268	135.5	31,300	18.6	$\frac{5}{8}$
		Bottom.	166,006	145.5	33,610	20.2	$\frac{19}{32}$
	A.L.P.	Top.	168,268	134	30,954	18.4	$\frac{19}{32}$
		Bottom.	166,006	149	34,419	20.7	$\frac{3}{4}$
Port, ...	H.P.	Top.	45,805	56	12,936	28.2	$\frac{17}{32}$
		Bottom.	43,543	61	14,091	32.3	$\frac{27}{32}$
	I.P.	Top.	125,123	120	27,720	22.1	$\frac{7}{16}$
		Bottom.	122,861	131.5	30,376	24.7	$\frac{3}{4}$
	F.L.P.	Top.	168,268	135	31,185	18.5	$\frac{9}{16}$
		Bottom.	166,006	153.5	35,458	21.3	$\frac{3}{4}$
	A.L.P.	Top.	168,268	130.5	30,145	17.9	$\frac{13}{32}$
		Bottom.	166,006	155	35,805	21.6	$\frac{25}{32}$

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ends and packings. Water and vacuum spaces are provided with all necessary drains.

Diameter of shell, inside, feet	7
Thickness of shell, inch	$\frac{3}{8}$
Length over heads, feet and inches	17- 1
Tubes, diameter, outside, inch	$\frac{5}{8}$
length between tube sheets, feet and inches	14- 6
as ordered, over all, feet and inches	14- $8\frac{1}{8}$
thickness, No. 18 B.W.G., inch	.048
spacing between centers, inch	$\frac{15}{16}$
number in each condenser	5,684
Cooling surface, both condensers, square feet	26,966

Main Air Pumps.—For each main engine there is a Blake, twin, vertical, beam, single-acting air pump, having two steam and two water cylinders. On the top of a lower suction base are the suction valves, eighteen in number for each pump barrel, and in the pump pistons and in the discharge chamber are the bucket and discharge valves respectively, eighteen each for each pump barrel. These valves are all uniform in size and construction, consisting of three bronze discs of graduated sizes, of which the top disc is plain except for the stud hole, while the other two discs have small holes within the rim of the overlying disc to allow the escape of water between the separate discs. The discs are held against their seats by a guard and spiral springs secured to the central stud. Examination of the suction and bucket valves is provided for by removable manhole covers which form a part of the pump barrel, the bucket being placed at the top and bottom positions respectively. Examination of the delivery valves is made through hand holes in the delivery chests, and these delivery valves have also a removable seating in contrast with the suction and bucket valves where the seating is a part of the valve plates.

Diameter of steam cylinders (2), inches	14
pump cylinder (2), inches	35
Stroke, inches	18
Diameter of piston rods, nickel-steel, inches	$2\frac{3}{4}$
pump rods, Tobin bronze, inches	$3\frac{1}{4}$
valves in bucket, head and foot, inches	$5\frac{1}{4}$
Number of valves (18 each bucket, head and foot), each barrel,	54

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Number of valves one complete pump, both barrels	108
Area through 18 valve openings, square inches	234
Lift of valves, possible under guards, inch	$\frac{21}{32}$
Discharge (area × lift), square inches	153.55

Main Circulating Pumps.—For each engine there is one centrifugal, double-inlet, circulating pump driven by a two-cylinder, compound, vertical, inverted engine. The specified capacity of each of these pumps is 12,000 gallons per minute when discharging from the bilges when running at not more than 266 revolutions per minute. Each pump draws from the sea, the main drain and from the engine-room bilge, and discharges to the main condenser, through its tubes and overboard when performing the regular duties, or directly overboard through the condenser by-pass, if desired, when in use on the bilges. The suction valves are so locked with inter-connecting gear that no two of them may be open at the same time.

Diameter of steam cylinders, inches	H.P., 11; L.P., 22
Stroke, inches	10
Diameter of pump runner, inches	45
Width of pump runner at periphery, inches	4
hub, inches	$10\frac{9}{16}$
Diameter of inlet nozzles (2), inches	$15\frac{1}{2}$
outlet nozzle, inches	21

Feed and Filter Tanks.—Abaft the main condenser in each engine room is located a combined feed and filter tank of about 5,330 gallons capacity. The after end of the tank, higher than the forward part, is arranged as a filter and grease extractor, and is in communication with the lower part of the tank over the top of a weir at the back. Filtering material is packed loosely in the three compartments of the filter section of the tank, and all discharges from main and from auxiliary air pumps are so led that the water must pass three times downward through the filter sections in succession before reaching the feed tank proper. Vapor pipes connect to the atmosphere from the top of the tanks, and all necessary gauge glasses, drains, overflow outlets and suction connections are provided, as well as bolted covers and manhole plates, for the proper access to all parts. The suction outlets to the hotwell pumps

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and to the cross connection from which the feed pumps take their suctions are protected from inrush of air when water is low by a valve controlled by a float.

Main Feed Pumps.—There are four main feed pumps, one at the starboard side of each of the four athwartship firerooms, of the Blake, single-cylinder, double-acting, plunger type. The suction for these pumps is from the feed tanks direct (from the cross-connecting pipe between the feed tanks) and from the main into which the hotwell pump discharges through the feed heaters, no other suctions being provided. The discharge from these pumps is to the main feed line only.

Auxiliary Feed Pumps.—The auxiliary feed pumps are duplicates of the main feed pumps and are placed on the opposite side of the firerooms, four in number. The suction for the auxiliary feed pumps is from the feed tanks (through cross connection as for main feed pumps), from the auxiliary drain, from the sea, from the floor of its own compartment and from the bottom blow pipes of all boilers in its own compartment. (It is possible, also, to divert the feed water after passing through the feed heaters to enable the auxiliary feed pumps to handle it.) The discharges of these pumps are: to the feed main for all boilers, to the fire main, overboard through sea valve in their own compartments and through special fittings for fireroom purposes.

Main and auxiliary feed pumps, four each, center-packed plungers:	
Diameter of steam cylinder, inches	14
water cylinder, inches	10
piston rods, inches	2¾
Stroke, inches	15
Valves, suction and delivery, 10 each per pump, diameter, inches	3¼
area of opening (1), square inches	5⅛

Feed-Water Heaters.—In each engine room is a feed-water heater of 950 square feet of heating surface, so arranged that the discharge of the hotwell pumps may be passed through the heater or be by-passed, the auxiliary exhaust steam being used to heat the feed water on its way through the heaters. The design provides that the heating steam is on the outside

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of the tubes, being distributed by baffles, which also serve as supporting plates for the tubes, while the water passes into the lower half of the tubes and leaves through the upper half by way of suitable nozzles on the inlet and outlet head. The pressure of the auxiliary exhaust steam for the heaters is regulated by relief valves on the connections from the exhaust line.

Diameter of shell, inside, feet and inches	2–11½
Thickness of shell, inch	¼
Length over heads, feet and inches	8– 9
Tubes, diameter, outside, inch	⅝
length between tube sheets, feet and inches	6–10
total, as ordered, feet and inches	6–11¾
thickness, No. 16 B.W.G., inch	.065
spacing between centers, inch	15/16
number in each heater	852
Heating surface, both heaters (950 each), square feet	1,900

Hotwell Pumps.—There is a hotwell pump in each engine room with suction connections for drawing water from the main feed tanks, from the reserve feed tanks in inner bottoms, from either main air-pump suction pipe, and from the fresh-water connections at the ship's side for taking in fresh water from lighters alongside. There are discharge connections through the feed heaters into the feed suction mains from which the main feed pumps draw, into the feed tanks, and into the reserve feed tanks.

Distiller Circulating Pump.—In the starboard engine room is a distiller circulating pump, a duplicate of the hotwell pumps, having a suction to the sea only, and discharging either through the distillers or by-passing them, into the fire main, the flushing system, or overboard. Relief valves are fitted on the discharge pipes to the distillers to guard against excessive pressure on the distiller coils. Hotwell (2) and distiller-circulating (1) pumps, Blake light-service piston type.

Diameter of steam cylinder, inches	12
water cylinder, inches	14
piston rods, inches	2¼
Stroke, inches	12
Valves, suction and delivery (18 each per pump), diameter, inches	3½
area of opening (1), square inches	5

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Fire and Bilge Pumps.—There is one fire and bilge pump in each engine room, Blake, vertical, simplex piston type, each having suctions for drawing water from the sea, the auxiliary drain, the drainage manifolds, the engine-room bilge and the crank pits. They discharge to the fire main and overboard.

Diameter of steam cylinders, inches	12
water cylinders, inches	10
piston rods, inches	2¼
Stroke, inches	12
Valves, suction and delivery (18 each per pump), diameter, inches	2¾
area of opening (1), square inches	2.57

Shaft Bilge Pumps.—On each main-engine shaft there is a bilge pump, worked by a crank pin on the end of the main shaft. These pumps have a capacity of about 390 cubic inches per stroke, and draw water from the engine-room bilges and from the crank pits, discharging overboard.

Auxiliary Condensers.—There is one auxiliary condenser in each engine room and one in the lower dynamo room, starboard side, of the type known as the Wheeler condenser, having a combined air and circulating pump attached to the condenser and located on the foundations under it.

	Engine rooms (2).	Dynamo room (1).
Diameter of shell, inside, inches	28	35¼
Thickness of shell, inch	⅜	⅜
Length over heads, feet and inches	7–10¾	9– 1½
Tubes, diameter, outside, inch	⅝	⅝
length between tube sheets, feet and inches	6– 8	7– 4
total, as ordered, feet and inches	6–10⅛	7– 6⅛
thickness, No. 18 B.W.G., inch	.049	.049
spacing between centers, inch	13/16	15/16
number in each condenser	552	1,001
Cooling surface, one condenser, square feet	601.4	1,200.9
Pump cylinders, steam, diameter, inches	8	12
circulating, diameter, inches	10	14
air, diameter, inches	10	14
piston rods (2), diameter, inches	1½	1¾
stroke, inches	12	12
Valves, air, suction and delivery, number each per pump	4	4
diameter, inches	5⅝	7
area of opening (1), square inches	13.6	21.5
circulating, suction and delivery, number each per pump	4	6
diameter, inches	5⅝	5⅝
area of opening (1), square inches	13.6	13.6

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Screw Propellers.—The propellers are of manganese-bronze, and are arranged to turn inboard when going ahead, the starboard propeller being left-hand and the port one right-hand. They are designed as true screws and are accurately balanced, with pitch adjustable from 20 feet 9 inches to 18 feet 9 inches, the designed pitch being 19 feet 9 inches. Each boss is accurately bored to fit the taper on the end of the shaft, is fitted with two feather keys, and is held by a nut screwed on the end of the shaft against the boss and locked in place. A cap of composition is fitted watertight over the end of the shaft and nut to fair the end of the boss and protect the shaft. The bolt recesses in blade flanges for securing the blades to the seatings in the boss are covered by watertight plates, set in flush with the flange surfaces. Blades, bosses, caps and plates are smoothed, finished and burnished.

Number of blades, each propeller	3
Diameter of propeller, feet and inches	16– 3
hub, feet and inches	4– 6
Length of hub, inches	42
Pitch of propellers, official trial, feet and inches	20– 6
Helicoidal area, each, square feet	80
Disc area, each, square feet	207.395
Projected area, each, square feet	64.5
Center of hub above lowest point of keel, feet	9.2
Top of propeller immersed (22 feet 6 inches draught), inches	62½

After the official trials of the *Charleston* the Bureau of Steam Engineering directed the contractors to make the spare propeller blades for this vessel to conform to the following dimensions: Diameter of propellers, 17 feet 6 inches; helicoidal area, 95 square feet; pitch, 19 feet 6 inches, mean. The form of the new blades is to be similar to that of the originals, and the same range of adjustability of pitch is required; that is, one foot above and below mean pitch.

Boilers.—There are sixteen Babcock & Wilcox boilers, arranged in four groups of four each, in four watertight compartments across the center line between the coal-bunker bulkheads. The two forward boilers of each group are located to starboard and port of the center line, respectively, and face

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the corresponding two after boilers of the same group, the athwartship fireroom for the group being between the forward and after pairs. There is a separate smoke pipe for each of the four groups of four boilers each, the vertical axis of the pipes being over the centers of the fireroom spaces for the four groups. The general type and arrangement of these boilers corresponds to those of the *West Virginia* and *Maryland*, of which a detailed description will be found in the illustrated article by Lieutenant R. K. Crank, U. S. Navy, on page 1223 *et seq.*, Vol. XVI, November issue, 1904, of the JOURNAL. The feature of the division of the furnaces of the boilers of those vessels by a water leg of four square-sectioned middle boxes and a 4-inch tube is not found in the boilers of the *Charleston*, where the number of sections making up the width of the boiler is smaller and the width of the furnace is consequently less.

Number of boilers	16
furnaces	16
Steam drum, inside diameter, inches	42
thickness, inch	$\frac{21}{32}$
Floor space, one boiler, length, feet and inches	10- 1
width, feet and inches	13- 8½
Grate, length, feet and inches	7- 0
width, feet and inches	12- 6
surface, one boiler, square feet	87.5
all boilers, square feet	1,400
Heating surface, one boiler, square feet	4,000
all boilers, square feet	64,000
Heating surface ÷ grate surface, ratio	45.71
Tubes, one boiler, 2-inch, No. 8 B.W.G., 9 feet 2 inches long	672
4-inch, No. 6 B.W.G., 9 feet 2 inches long	23
8 feet 11 inches long	2
7 feet 3 inches long	23
Elements, one boiler (21 intermediate, 2 sides)	23
Weight of water, one boiler, at steaming level, pounds	13,785
complete, pounds	80,796
Working pressure, on trials, pounds, per square inch	300
Air pressure on trials, inches of water column	2
Safety valves (1 triple-type each boiler), total	48
diameter of each valve, inches	3
set to blow at, pounds	300
Sentinel valves (1 each boiler), diameter, inch	½

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Sentinel valves set to blow at, pounds	295
Smoke pipes (4), diameter, outside, feet and inches	8– $5\frac{9}{16}$
inside, feet	8
area, each, square feet	50.2655
total, square feet	201.062
height above grates, feet and inches	92–$10\frac{1}{2}$
Escape pipes (4), diameter, inches, after pipe	10
three others	9

Forced-Draft Blowers.—There are eight forced-draft blowers of the Sturtevant type located in recesses in the uptakes and above the protective deck. The arrangement and location of these blowers is such that two blowers discharge into each of the four firerooms, taking their suction from the air spaces around the base of the smoke pipe of the boilers they supply, suitable airtight shutters being fitted over boiler-hatch openings to make the firerooms tight. All blowers are readily accessible from the berth-deck compartments and are well protected from the dust and heat of the firerooms, so that they may be well cared for under forced-draft conditions and may be expected to be far more efficient and durable than in the firerooms. By opening the access doors to the blower engines while in operation the blowers become very serviceable as additional ventilation to the lower-deck spaces by the air drawn into the casings from these spaces.

Forced-draft blowers (8), two-cylinder, double, vertical, enclosed :	
Steam cylinders (2), diameter, inches	7
Stroke, inches	5
Diameter of piston rods, inches	$1\frac{5}{16}$
fan, inches	66
Area of induction nozzle, square inches	1,410.3
eduction nozzle, square inches	1,755

General Workshop.—The following tools are provided and installed in the general workshop on the port side of the berth deck abreast of the engine-hatch enclosure:

One lathe, screw-cutting, back-geared, gap, swinging 30 inches over the ways and taking 10 feet between centers;

One lathe, screw-cutting, back-geared, 14-inch, taking 4 feet between centers;

One column-shaping machine, 16-inch stroke, 24-inch traverse;

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One drilling machine, back-geared, to drill up to 1½ inches, 14 inches from edge of work;
One sensitive drill, 16-inch;
One milling machine, universal, No. 1;
One combined punch and shears, hand, 6-inch blades, with capacity for shearing ¾-inch round iron, ⅜-inch steel plate, and punching ⅜-inch holes in ⅜-inch mild-steel plates;
One emery grinder, two wheels, 12-inch diameter and 2-inch face;
One grindstone, 30-inch;
Four bench vises are mounted on two vise benches convenient for hand work, two other vices being mounted on small benches in the engine rooms.

The above power tools are driven by a system of overhead line and counter shafting belted from a 10-H.P. electric motor in the forward outboard corner of the workshop.

Ash Hoists.—Four fireroom ventilators, one in each athwartship fireroom, alternately starboard and port from forward, are fitted with sets of angle-iron guide strips, which guide the ash buckets in their ascent from the fireroom to the main-deck level. These guides are so fitted that they do not interfere with the placing of the armor bars in the protective-deck openings when the ash hoists are not in use. The hoisting is done by four double-cylinder, vertical, reversible steam engines of the Williamson Brothers' servo-motor type, located within the fireroom hatch enclosures at the main-deck level and worked by means of hand wheels and geared shafting from the hoisting station. The engine cylinders (two) are 4½ inches in diameter by 4½ inches stroke; the cranks are set at right angles, and safety gears are provided to prevent overwinding and to stop the buckets when they reach the fireroom floor. From the four hoisting stations the ash buckets are carried by trolley tracks to their corresponding ash chute direct, these chutes being located at frame spaces 48–49 and 58–59, starboard and port, or, by cross-connecting trolley tracks and switches, either of the forward two stations may dump in either of the forward chutes, and either of the after two stations

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may dump in either of the after chutes. For dumping into a lighter alongside, the buckets are carried on trucks through the forward doors of the superstructure and dumped in portable chutes secured to the water-ways.

Distilling Apparatus.—There are four evaporators, of the usual Navy type, of a combined capacity of 21,000 gallons per day, and three distillers of the horizontal-tube type, in which the steam passes through the tubes and the cooling water around them, having a combined capacity of 21,000 gallons of potable water per day. The distillers are located in the fireroom-hatch enclosure, forward of the after smoke pipe, at the level of the gun deck, the evaporator room being on the berth deck below them. Aside from the distiller circulating pump before mentioned, there is a connection from a riser to the fire main, from which the circulating water may be supplied to the distillers, using one of the pumps which discharge to the fire main.

Evaporators (4).....3 port side athwartships, 1 starboard side fore-and-aft.	
Shells, steel, diameter, inside, feet.....	5
*thickness, inch.....	(2) $\frac{3}{8}$, (2) $\frac{1}{4}$
†length over heads and braces, feet and inches.....	5- $8\frac{3}{4}$
†total over all, including heads and stiffeners, feet and inches.....	6- $8\frac{1}{2}$
Heads, steel, thickness, inch.....	$\frac{1}{2}$
Tube sheets, composition, thickness, inch.....	$\frac{3}{4}$
Tubes, brass, No. 12 B.W.G., one evaporator, number.....	96
outside diameter, inches.....	2
†length between tube sheets, feet and inches.....	5- $3\frac{1}{2}$
†total, as ordered, feet and inches.....	5- $5\frac{1}{4}$
†Tube-heating surface, one evaporator, square feet.....	266
†Capacity, one evaporator, per day, gallons.....	5,250
Dry-pipe, brass, diameter, inside, inches.....	5
slots ($\frac{1}{4}$ inch × 130 degrees arc, transverse), number..	20
area ($\frac{1}{4}$ × 5.672 × 20), square inches.....	14.18
Steam connection to coil, diameter, inches.....	$2\frac{1}{2}$
Drain connection to trap, diameter, inches.....	$1\frac{1}{2}$
Distillers (3).....	horizontal.
Shells, cast iron, diameter, inside, inches.....	$19\frac{1}{2}$

*Original design for high- and low-pressure evaporators in double effect not changed though single-effect system is now used.

†After port evaporator shortened 4 inches, shell and tubes, to allow withdrawal of coils in space allotted.

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Shells, thickness, inch	$\frac{7}{16}$
length over all, feet and inches	5– $8\frac{7}{8}$
Tubes, brass, No. 16 B.W.G., one distiller, number	192
outside diameter, inch	$\frac{5}{8}$
length between tube sheets, feet and inches	4– $5\frac{1}{2}$
total, as ordered, feet and inches	4– 7
Tube cooling-surface, one distiller, square feet	140
Capacity, one distiller, per day, potable water, gallons	7,000
Circulating water, inlet and outlet, diameter, inches	9
Steam inlet, diameter, inches	9
Fresh-water outlet, diameter, inches	$3\frac{1}{2}$

Refrigerating Plant.—One horizontal, two-ton, Allen, dense-air, ice machine is installed in its compartment on the starboard side of the berth deck between frames 73 and $75\frac{1}{3}$, from which refrigerating piping and the necessary returns are taken to the refrigerating rooms, the after scuttle butt and the ice-making tank. The refrigerating rooms are four in number—captain's, junior and warrant officers', wardroom officers', and crew's—with a common air lock from which all open, and are located between frames 68 and 73 on the starboard side of the berth deck. The after scuttle butt is located just forward of frame 73 on the gun deck, starboard side, over the after end of the refrigerating rooms, the forward scuttle butt not being piped with cooling coil. The ice-making tank is located in the after inboard corner of the ice-machine room. The general location of the plant is unfortunate so far as efficiency is concerned owing to the high natural temperature of that portion of the ship and the close proximity of the engines and boilers. As the ice machine is in one unit this will also operate to lower the continued efficiency of the plant through the necessity of long-continued running of the machine with but little chance for overhaul. Another obstacle to equal efficiency of all the rooms lies in the fact that the piping is in combination for the captain's and the wardroom officers' spaces, and for the junior and warrant officers' and the crew's spaces, so that the first mentioned of each of the two combinations must receive the full cooling effect before the second mentioned, and the circulation must be kept up on the whole combination even after one of the two spaces is cooled to the desired

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degree. The ice-making tank may be used (a) alone, (b) with rooms, (c) with rooms and scuttle butt, (d) with scuttle butt, or (e) it may be cut out when not required. From the foregoing, with proper substitution of names, may be seen the combinations of uses possible with the plant.

Cylinder, steam, diameter, inches	$9\frac{3}{4}$
compressor, diameter, inches	7
expander, diameter, inches	$5\frac{3}{4}$
circulating pump, diameter, inches	$3\frac{1}{2}$
primer pump, diameter, inches	2
Stroke of crank, inches	13
Piston rods (3), diameter, inches	$1\frac{5}{8}$
length, over all, feet and inches	2– $8\frac{1}{4}$
Cooling coil, face to face of elbows, length, feet and inches	4– $2\frac{13}{16}$
center to center of ends of coil, length, feet and inches	3– $9\frac{5}{16}$
complete spiral, turns, number	$18\frac{1}{2}$
diameter, mean, inches	$12\frac{3}{4}$
tubing, diameter, outside, inches	2
inside, inches	$1\frac{3}{4}$

Anchor Windlass.—The anchor windlass is of the Hyde type, designed for handling four chains, and is located in a pocket between the main and gun decks. Its main worm wheel on the wildcat shaft is driven by a vertical shaft with a worm pinion at its upper end and a bevel gear at its lower end, the latter bevel gear meshing with a similar gear on the after end of the engine shaft. Suitable thrust bearings are provided to take the end thrust of the engine shaft and of the vertical worm shaft, the latter having also a step and a top bearing. The windlass engine is located on the gun deck just forward of the windlass pocket and is of the two-cylinder, vertical, inverted, reversible type, having two 15-inch cylinders of 14-inch stroke.

Steering Engine.—The steering engine is of the Williamson Brothers' design, arranged for steam and hand control, the usual steam-steering wheels being fitted on the flying bridge, forward bridge, and in the conning tower. Electric and direct-acting indicators are installed showing the position of the helm, and an electric transmitter communicates to the hand-steering station. The engine is two-cylinder, horizontal, direct-

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acting; diameter of cylinders, 15 inches; stroke, 12 inches; and diameter of piston rods $2\frac{1}{16}$ inches.

Electric Plant.—The electric plant of this vessel consists of two 100-kilowatt and three 50-kilowatt generating sets of 125 volts pressure at the terminals; about one thousand electric fixtures, complete, with all necessary incandescent lamps; twelve enclosed arc lamps of about 3 ampères capacity in engine andfirerooms; six 30-inch searchlights; two truck lights; two electric night-signaling sets, complete; two diving lanterns with 150-candlepower incandescent lamps; six portable electric ventilating sets; forty $\frac{1}{12}$-horsepower desk and bracket fans; six $\frac{1}{8}$-horsepower bracket fans; one electric whistle operator; all necessary panel and switch boards for lighting and power; with all wiring, accessories, fixtures, lamps, instruments, tools, spare parts and stores necessary for the proper manipulation, test and repair of the plant. The generating sets are secured to a common bedplate, the engine being direct-connected to the dynamo. The engines are of the vertical, cross-compound, General Electric Company's type, all working parts enclosed and lubricated under pressure, the introduction of oil into the cylinders being guarded against by a top plate on the enclosing covering, having soft-packed stuffing boxes for the rods and valve stems to work in. The dynamos are of the direct-current, compound-wound, multipolar type, and the wiring is arranged on the two-wire feed system.

50-kw. dynamo engines, number	3
Diameter of cylinders, H.P., inches	$7\frac{1}{2}$
L.P., inches	14
Stroke, inches	8
Diameter of piston rods, H.P., inches	$1\frac{3}{4}$
L.P., inches	$1\frac{3}{4}$
100-kw. dynamo engines, number	2
Diameter of cylinders, H.P., inches	10
L.P., inches	18
Stroke, inches	10
Diameter of piston rods, H.P., inches	$2\frac{1}{4}$
L.P., inches	$2\frac{1}{4}$

Motors.—In connection with the electric plant, and for the operation of the various auxiliaries outside the engine and

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fireroom spaces (all of which, with the exception of the forced-draft blowers, ice machine, evaporator pumps, windlass and steering gear, are operated by electricity), there are installed the following motors:

Use.	No.	H.P. Each.	H.P. Total.	Winding	Current. (amp.)	Location.	Makers and remarks.
Ventilation.							
70-in. S. P. blower.	2	...	...	Shunt.	64	Protective deck.	B. F. Sturtevant Co.
60-in. S. P. blower.	4	...	...	Shunt.	64	Eng. room and forward hold.	B. F. Sturtevant Co.
50-in. S. P. blower.	2	...	...	Shunt.	36	Berth and gun decks.	B. F. Sturtevant Co.
45-in. S. P. blower.	3	...	...	Shunt.	26	Prot., berth and gun decks.	B. F. Sturtevant Co.
35-in. S. P. blower.	6	...	...	Shunt.	15	Prot., berth and gun decks, power platform aft, and steering eng. rooms.	B. F. Sturtevant Co.
No. 4 Monogram ..	6	...	...	Shunt.	10	Berth and gun decks and upper platform.	B. F. Sturtevant Co.
No. 3 Monogram..	3	...	...	Shunt.	6.80	Gun deck.	B. F. Sturtevant Co.
No. 1 Monogram...	1	...	...	Shunt.	1.26	Berth deck.	B. F. Sturtevant Co.
Boat cranes.							
CB 25.....	2	35	70	Series.	240	Boat cranes.	Gen. Electric Co.
CB 31.....	6	20	120	Series.	140	Bridge deck.	Gen. Electric Co.
Deck winches.							
Type M.....	7	25	175	*Series.	175	Main and upper deck.	Thresher Elec. Co.
Ammunition hoists...	27	2.5	67.5	Shunt.	19	Hold, amm'n passage and berth deck.	Gen. Electric Co.
Ammunition winches (whip hoists.)	3	3.5	10.5	Shunt.	26	Upper deck and bridges.	Gen. Electric Co.
Pumps, fresh water..	2	2	4	Shunt.	14.7	For'd hold and steering eng. platform.	Holtzer Cabot Co.
Sanitary.....	1	½	½	Shunt.	3.9	Engine room.	Gen. Electric Co.
General workshop....	1	10	10	Shunt.	30	Gen. workshop.	Gen. Electric Co.
Laundry.....	1	6	6	Shunt.	42	Laundry.	Gen. Electric Co.
Bread mixer.....	1	2	2	Shunt.	14	Bakery.	Gen. Electric Co.

*Shunt winding sufficient to control speed at no load.

VENTILATION.

All the spaces below the gun deck, and the officers' and crew's closets on the gun deck, are provided with artificial ventilation. There are fifteen systems (or units) of ventilation, and these are grouped into four methods, viz:

The Supply and Exhaust Method.—This method is applied to all compartments where it is difficult both to supply the fresh air and to exhaust the foul by natural convection. Under this head come the dynamo rooms and the steering-engine room.

The Supply Method.—Air is supplied by forced ventilation to all compartments below the gun deck and between the

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'thwartship armor bulkheads, and also, where necessary, as in the magazines, exhaust ducts are provided to allow the escape of the foul air.

The Exhaust Method.—Where there is a natural inlet of fresh air through hatches and air ports, as in the case of living quarters aft on the berth deck, the foul air is exhausted by fans.

The Convection Method.—This is applied to the bunkers and firerooms where there is a natural flow of heated air upward through hatches and pipes provided for that purpose, and ventilators and supply pipes are arranged to carry the fresh air to supply the need.

The engine rooms are ventilated by four electric blowers, locted two on and two just below the protective deck. These blowers supply fresh air from ventilators to the four main ducts leading inboard and outboard in each engine room. From there ducts branches are led to the various points in the engine rooms where cool, fresh air is most needed. The preceding table of motors gives the number, type and location of the different ventilation blowers.

STANDARDIZATION OF SCREWS.

It having been decided to try the vessel according to the standardized screw method, progressive runs over the measured mile at Provincetown, Mass., were made and the data recorded in Tables I and II was obtained. From this data the curves on plates I and II were plotted.

It was determined that 143.081 revolutions per minute of the main engines would be required to give the contract speed of 22 knots per hour.

The standardization runs were commenced at 8·31 A. M. on June 28th and completed at 2·15 P. M. The draught and corresponding displacement at the beginning and end of the runs were as follows:

	Beginning.	*End.*
Draught, forward, feet and inches	22- 6	22-1
aft, feet and inches	22-10	22-6¾
mean, feet and inches	22- 8	22-3⅞
Displacement, tons	9,800	9,620

The coal used was New River, run of the mine.

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Table I.—U. S. S. *CHARLESTON*.

Data from Progressive Runs over Measured-Mile Course off Provincetown, June 28, 1905.

Number and direction of run.	Time on course.	Revolutions of main engines. To make one knot. Starboard.	Revolutions of main engines. To make one knot. Port.	Revolutions of main engines. Per minute. Starboard.	Revolutions of main engines. Per minute. Port.	Revolutions of main engines. Per minute. Mean.	Speed in knots per hour.	Indicated horsepower. Starboard engine. H.P. cylinder.	Starboard engine. I.P. cylinder.	Starboard engine. F.L.P. cylinder.	Starboard engine. A.L.P. cylinder.	Starboard engine. Total.	Port engine. H.P. cylinder.	Port engine. I.P. cylinder.	Port engine. F.L.P. cylinder.	Port engine. A.L.P. cylinder.	Port engine. Total.	Total both main engines.	Auxiliaries in operation (estimated).	Total all machinery.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
	m. s.																			
1-N	4-38.4	333.3	331.3	71.8	71.4	71.6	12.931	583.37	472.2	*148	169.70	1,373.27	580.06	469.5	213.97	244.09	1,507.52	2,880.89	...	...
2-S	4-28	361.8	355.5	81	79.6	80.3	13.433	790.7	659.5	*247.78	259.82	1,957 8	1,023.8	590.6	275.43	288.86	2,178.69	4,136.49	...	...
3-N	3-54	337.7	339.6	86.6	87.1	86.2	15.385	1,045.7	859.7	*285.65	321.64	2,512.59	1,071.1	829	349.14	393.35	2,642.48	5,155.07	...	...
4-S	3-43.07	376	370.8	101.1	99.7	100 4	16.136	1,688.5	1,367.7	*490.8	563.24	4,110.24	1,609.4	1,342.9	542.9	622.95	4,118.1	8,228.34	...	...
5-N	3-14.93	348.1	343.1	107.1	105.6	106.4	18.471	2,066.8	1,702.1	542.42	786.57	5,097.83	2,037.8	1,663.4	561.5	784.4	5,047.1	10,144.93	...	...
6-S	3-19.6	382.8	380.4	115.1	114.3	114.7	18.036	2,441.95	2,068.9	1,190.2	1,000.35	6,701.8	2,438.6	1,987.1	844.5	1,037.5	6,307.72	13,009.52	...	...
7-N	2-55.9	357.7	350.5	122	119.5	120.8	20.466	2,960	2,674.3	965.96	1,503.52	8,103.88	2,860	2,564.3	1,200.8	1,291.6	7,916.7	16,020.58	...	...
8-S	3- 2.5	400.7	400.8	131.8	131.8	131.8	19.726	3,604.15	3,456.1	1,576.4	1,734 23	10,370.88	3,442.1	3,486.4	1,579.3	1,673.8	10,181.6	20,552.48	...	...
9-N	2-52.2	393.3	322.5	137	112.4	124.7	20.906	3,826.5	4,590.4	2,110.5	2,283.95	12,811.35	...	...	...	...	...	...	...	...
10-S	2-56.66	401.6	399.6	136.4	135.7	136.5	20.873	3,917.45	4,143.2	1,850.25	1,991.85	11,902.75	3,875.4	3,629.9	1,816	1,913.4	11,234.73	23,137.48	...	...
11-N	2-43.66	376	374.8	137.8	137.4	137.6	21.99	3,327.8	4,470.6	2,041.4	2,140.3	11,980.1	3,083.4	4,841.9	2,152	1,885.2	11,962.5	23,942.6	...	...
12-S	2-50.7	397	399.1	139.5	140.3	139.8	21.089	3,463.3	4,630.5	2,060.65	2,215.75	12,368.2	2,960.3	4,876.1	2,279.45	2,451.15	12,567	24,935.2	...	...
13-N	2-41.9	384.9	384.5	142.7	142.5	142 5	22.236	3,156.3	5,451.6	2,321.1	2,541.55	13,470.55	2,911	5,355.4	2,466.1	2,676.6	13,209.1	26,679.65	...	...
14-S	2-46.1	395.9	394.6	143	142.5	142.8	21.67	2,985.5	5,445.2	2,444.4	2,559.05	13,434.15	2,895.4	5,088.5	2,412.45	2,671.1	13,067.35	26,501.5	...	...

* Defective cards interpolated from port engine.

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Table II.—U. S. S. *CHARLESTON.*

Data from Progressive Runs over Measured-Mile Course off Provincetown, Mass., June 28, 1905.

Number of run.	Time at beginning of run (approximate).	Which engine.	Steam pressures.					Mean effective pressures.					Vacuum, inches.	Cut off in cylinders from beginning.			Temperatures.							Revolutions or double strokes.								Air pressure, inches water.	Sanitary pump.
			In boilers above atmosphere.	At engine above atmosphere.	H.P. steam chest, gauge.	In first receiver, absolute.	In second receiver, absolute.	H.P. cylinder.	I.P. cylinder.	Forward L.P. cylinder.	After L.P. cylinder.	Referred to low pressure.		H.P.	I.P.	L.P.	Outside air.	Engine-room working platform.	Fireroom working level.	Injection.	Discharge.	Hotwell.	Feed.	Circulating pump.	Main feed pumps.	Aux. feed pumps.	Hotwell pump.	Main air pump.	Bilge pump.	Aux. cond'r pump.	Fireroom blowers.		
1	...	S.	...	282	55	22	11.5	36	10.5	0	2.8	10.11	28.7	Full gear.	Full gear.	Full gear.	70	75	...	60	92	150	150	120	...	...	18	20	39	10	225	...	20
		P.		270	55	20	7.75	...	10.5	3.55	4.05	...	28.5					91	...	55	98	122	...	98	7	...	12	18	35	59		...	...
2	...	S.	...	250	73	27	13	43.25	13	.7	3.8	12.86	28.5				70	85	...	60	88	140	140	120	...	...	18	20	39	10	255	...	20
		P.		245	70	25	8	57	11.85	4.1	4.3	16.22	28.5					91	...	58	88	110	...	97	18	...	11	19	38	60		...	...
3	...	S.	...	254	97	33	15	53.5	16.85	1.85	4 4	16.16	29				68	86	...	60	94	120	120	116	...	...	18	20	39	10	270	...	20
		P.		248	96	34	12	54.5	15.2	4.75	5.35	17.98	28.25					98	...	58	92	98	...	95	18	...	13	19	42	60		...	...
4	...	S.	...	270	130	43	12	74	21.6	4.25	6.6	23.34	29.5				64	87	...	60	96	115	160	116	...	...	17	20	39	10	281	...	20
		P.		270	152.5	37.5	15.5	71.5	21.5	6.45	7.4	19.46	27.5					94	...	59	100	84	...	112	11.5	...	10	18	40	22		...	...
5	...	S.	...	292	157	52	19	85 5	25.4	6	8.7	28.21	28.7				64	88	...	60	102	92	150	116	...	...	18	20	39	10	295	...	20
		P.		290	162.5	47.5	18.5	85.5	25.15	6.3	8.8	28.31	26.75					86	...	58	118	85	...	114	15	...	13	18	38	31		...	...
6	...	S.	...	262	180	57	24	94	28.7	12.25	10.3	34.49	28.5				61	89	...	60	110	88	142	120	...	...	30	20	38	10	295	...	20
		P.		270	187	57.5	23.5	94.5	27.75	8.75	10.75	32.68	26.75					88	...	58	116	86	...	135	18.5	...	7	19	41	41		...	...
7	...	S.	...	295	224	75	29	107.5	35	18.75	14.6	39 35	27				70	89	...	60	115	95	132	120	...	...	30	20	38	8	335	...	20
		P.		272	215	65	27	106	34.25	11.9	12.8	39.23	25.75					84	...	58	109	90	...	135	18.5	...	13	20	41	44		...	...
8	...	S.	...	240	235	87	33	125.25	41.9	14.18	15.6	46.65	28				62	89	...	60	112	92	158	122	...	...	30	20	40	8	375	...	18
		P.		240	237	83	33	115.75	42.25	14.2	15.05	45.78	25.50					90	...	58	118	90	...	147	25	...	31	21	38	38		...	...
9	...	S.	...	...	...	...	...	123.75	53.5	18.25	19.75	35.4	...				...	...	...	...	...	...	...	180	...	...	40	22	40	6	380	...	18
		P.		...	...	...	...	...	...	...	...	...	...					...	...	...	...	...	...	183	...	...	26	23	41	30		...	...
10	...	S.	...	250	247	102	38	127.25	48.5	16 7	17.3	51.6	28				66	88	...	60	102	98	150	180	...	...	40	23	40	6	495	...	16
		P.		250	247	95	36	126.5	42.7	15.85	16 7	49.03	26					92	...	59	112	89	...	178	27	...	22	21	44	38		...	...
11	...	S.	...	252	250	109	41	107	51.8	17.55	18 4	51.5	27.5				71	88	...	60	120	100	160	180	...	...	40	24	40	6	505	...	16
		P.		255	252	96	37	99.4	56.25	18.55	16.25	46.56	27					91	...	59	116	110	...	171	27.5	...	23	21	37	39		...	...
12	...	S.	...	255	252	112	41	110	53	17.5	18.8	52.52	27				64	87	...	60	120	104	155	184	...	...	44	23	44	6	500	...	16
		P.		255	252	110	42	93.5	55.5	19.25	20.7	53.07	25.5					91	...	59	118	112	...	169	28.5	...	18	21	41	34		...	...
13	...	S.	...	260	255	127	45	98	61	19.27	21.1	55.87	27				64	86	...	60	122	102	162	200	...	...	44	24	48	6	540	...	16
		P.		260	257	115	44	90 5	60	20.5	22.25	55.74	26					92	...	58	118	112	...	173	30	...	29	19	43	30		...	...
14	...	S.	...	250	250	120	45	92.5	60.8	20.25	21.2	55.65	27.5				68	86	...	60	120	100	155	200	...	...	44	24	48	6	445	...	16
		P.		260	257	115	44	90	57	20.05	22.2	54.3	26					91	...	58	118	111	...	178	32	...	36	21	72	34		...	...

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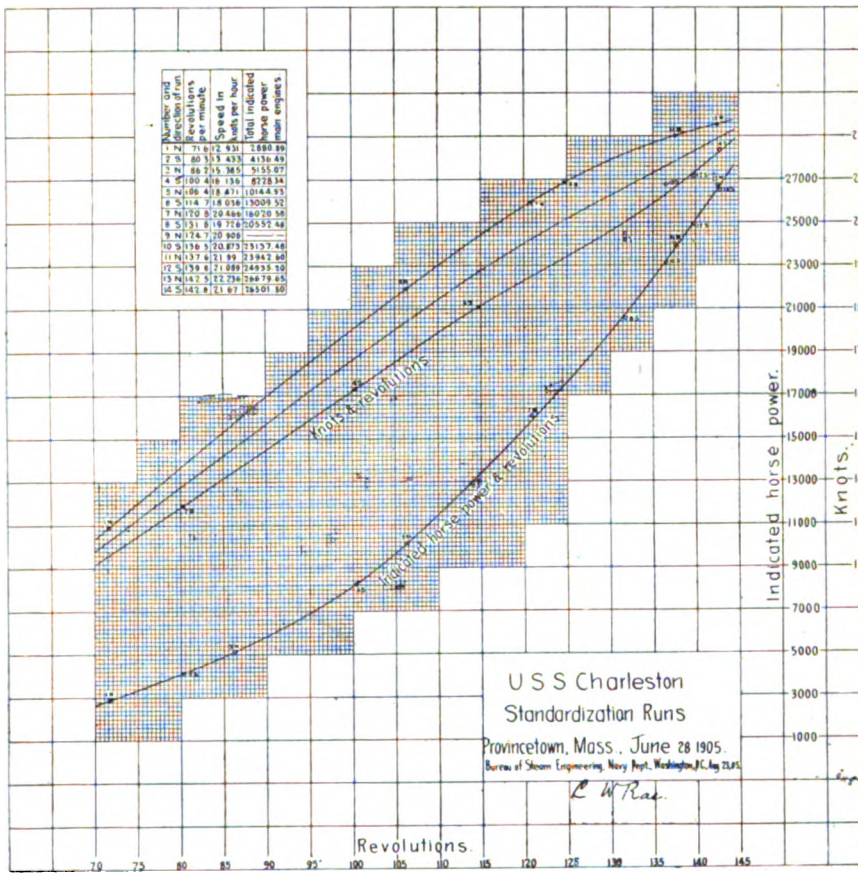

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OFFICIAL FOUR-HOURS' TRIAL.

At 7·30 A. M. on June 29, the *Charleston* stood out to sea for the official four-hours' trial which commenced at 8·53 A.M. The weather was clear and pleasant with a light S.W. breeze and smooth sea.

The draught and displacement at the beginning of the trial were as follows:

Draught, forward, feet and inches	22- 3¼
aft, feet and inches	22-11½
mean, feet and inches	22- 7⅝
Allowance for anchor, etc., inch	⅝
Mean draught, feet and inches	22- 7¼
Displacement, tons	9,775

The estimated displacement at end of trial was 9,587.8 tons. A synopsis of the data obtained follows.

PERFORMANCE.—FOUR-HOURS' OFFICIAL TRIAL.

Steam Pressures. (*Average of one-half hourly observations.*)

	Starboard.		*Port.*
Steam pressure at boilers (per gauge), pounds		286.7	
H.P. steam chest (gauge), pounds	252.8		254.2
1st receiver (absolute), pounds	125.7		122.9
2d receiver (absolute), pounds	47.4		45.4
Vacuum in condensers, inches of mercury	27.3		26.42

Revolutions, or Double Strokes, per Minute. (*Average of one-half hourly observations.*)

Main engines (mean for trial)	143.2		143.59
average for both		143.395	
Pumps, main air	24.8		23.9
circulating	192.2		181.8
feed (forward and after)		32.4	
hotwell	45.8		35.9
bilge	44.7		47.3
Blower engines (average)		529.5	
Speed of ship, in knots per hour		22.04	
Slip of propeller, in per cent. of its own speed, on mean pitch and average speed	23.96		24.16
Air pressure of firerooms, inches of water		2.48	

Mean Effective Pressures in Cylinders, in pounds per square inch. (*Average of cards taken at half-hourly periods.*)

Main engines, H.P. cylinder	91.463	83
I.P. cylinder	58.823	59.875
F.L.P. cylinder	21.954	21.438
A.L.P. cylinder	21.02	22.913
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	55.54	55.483

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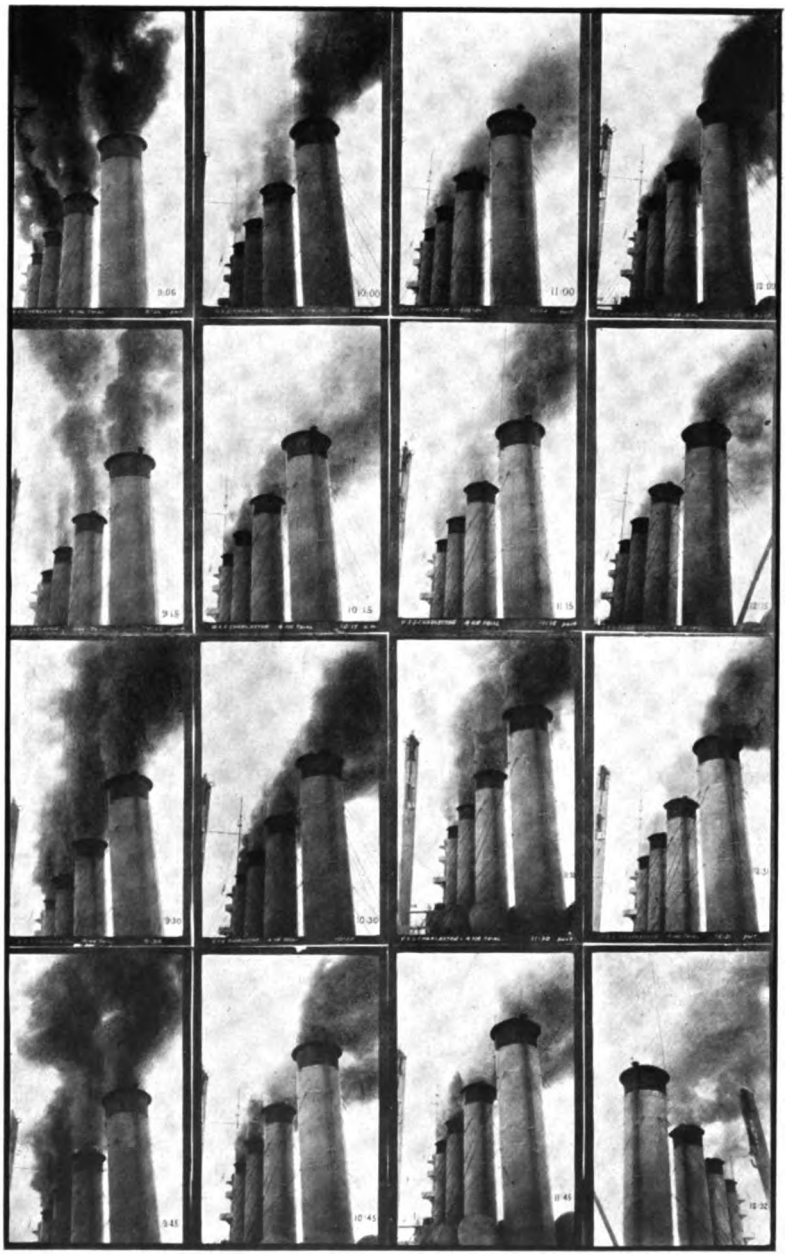

PHOTOGRAPHS OF SMOKE TAKEN DURING TRIAL, U. S. S. "CHARLESTON."

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INDICATED HORSEPOWER.

	Starboard.	*Port.*
Main engines, H.P. cylinder	2,956.1	2,689.9
I.P. cylinder	5,275.6	5,384.5
F.L.P. cylinder	2,653.7	2,598.4
A.L.P. cylinder	2,540.8	2,777.1
Total	13,426.2	13,449.9
both engines	26,876.1	

AUXILIARIES.

Air pump, main	11.15	11.62
Circulating pumps, main	71.5	67.63
Feed pumps (main)	130.44	
Hotwell pumps	31.7	
Bilge pumps	7.44	
Auxiliary condenser pumps	6	
Blower engines (forced draft)	216	
Steering engine (estimated)	5	
Dynamo engine (one in operation)	72	
Total	630.48	
Main engines, air, circulating, hotwell and feed pumps	27,200.14	
Total all machinery in operation	27,506.58	

COAL.

Kind and quality	New River, hand-picked, excellent.
Pounds per hour	57,265

DEDUCED DATA.

I.H.P. (total) per square foot of grate surface	19.65
heating surface	.4298
Main engines, air, circulating, hotwell and feed pumps per square foot of grate surface	19.43
Main engines, air, circulating, hotwell and feed pumps per square foot of heating surface	.425
Pounds of coal per I.H.P. per hour (main engines, air, circulating, hotwell and feed pumps)	2.11
Pounds of coal per I.H.P. per hour, all machinery in operation	2.08
Pounds of coal per square foot of grate surface	40.9
heating surface	.895
Cooling surface (main condenser), square feet per (total) I.H.P.	.98
Heating surface, square feet per (total) I.H.P.	2.327

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COAL CONSUMPTION OF U. S. CRUISERS *CINCINNATI*, *RALEIGH*, *NEW ORLEANS* AND *ALBANY*.

By Commander W. W. White, U. S. Navy (Retired), Member.

The accompanying table has been compiled from the results of a run, in squadron, of the *Cincinnati*, *Raleigh*, *New Orleans* and *Albany*, in December of 1903. A short time prior to the trip all ships had been docked and their bottoms cleaned. The coal used (Pocahontas) was from the same Government pile at Yokohama.

The principal data of these ships is as follows:

——	*Cincinnati* and *Raleigh* (steel).	*New Orleans* and *Albany* (steel, wood-sheathed).
HULL.		
Length on L. W. L. (normal), feet	300	346
Extreme breadth, feet and inches	42	43- 9
Mean draught (normal), feet	18	18
Displacement (normal), tons	3,213	3,769
MACHINERY.		
Engines	4-cylinder, triple-expansion.	3-cylinder, triple-expansion.
Diameter of cylinders, inches	24 × 44½ × 57 × 57	31 × 46 × 70
Stroke, inches	33	30
Boilers	8, B. & W.	4, double-ended (Scotch).
Designed pressure	225	160
Grate surface, total, square feet	506	474
Heating surface, square feet	21,016	13,768

The auxiliary machinery in operation for all ships during the run was very closely the same, with the following exceptions: On the *Cincinnati*, all fresh water needed both for ship's use, and to supply deficiency of boiler feed, was made, as required, by the evaporating plant. Other ships distilled water for ship's use, but to replenish feed water drew frequently

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UNITED STATES ASIATIC FLEET-CRUISER SQUADRON.

Performance data, Yokohama, Japan, to Honolulu, H. I. = 14 days' run. Data recorded at 10 A. M. daily.

December 3–16, 1903.	Date.	4	5	6	7	8	9	10	11	12	†12	13	14	15	16	Total.
Distance made good, knots.	* By observation..	‡225.3	224.3	‡227.8	228.7	244	232.8	241.8	252	245.8	‡242.3	‡292.4	310.3	301.3	‡289 5	3,558.3
	* By log.....	227.6	229.6	230.1	225.1	248.3	241.4	242.2	244.2	245.8	244.7	295.4	303.3	311.1	292.4	3,581.2
	Average per hr.	9.4	9.7	9.6	9.6	10.3	9.8	10.2	10.6	10.3	10.2	12.3	13	12.7	12.2	...
Average revolutions per minute.	*Cincinnati*.....	70.2	68.8	70.1	68.4	73.9	73.4	73.7	73.8	74.3	74.4	89	92.5	94	90	...
	Raleigh.....	72.9	72.4	75.8	72.6	78.6	78.7	78.6	78.6	79.7	79.5	96	99.4	101.4	97.8	...
	New Orleans....	73	71.5	73.7	72	76.5	76.9	77	77	76.4	77	91.6	95	97.2	94.2	...
	Albany.....	74.7	74.6	77.2	74.9	79.5	77.9	79.9	79.3	79.7	79.8	95.2	98.3	100.5	97.3	...
Coal per day, tons.	*Cincinnati*.....	29.2	29.4	29.4	28.1	31.9	35.7	31	30.3	30.5	27.8	34.7	34.1	37.4	32.5	442
	Raleigh.....	36	35.3	28	24.6	30.2	28.8	30.1	31.6	30.5	30.6	45.3	48.6	50.6	52.4	502.6
	New Orleans....	30.5	31.7	32.8	31.8	32.9	33.2	34.5	35.1	32.6	33.1	44.2	44.7	47.8	47.7	512.6
	Albany.....	37.6	35.8	35.4	33.4	34.2	35.2	35.4	34.7	35.6	35	43.9	47.4	50.5	54.9	549

Draught, feet, inches, and Displacement, tons.	*Cincinnati.*	*Raleigh.*	*New Orleans.*	*Albany.*
December 3.				
Forward.....	19	18	18– 6	19– 4
Aft.....	20– 6	20	19– 6	19–10
Displacement ...	3,655	3,450	4,030	4,190
December 16.				
Forward.....	15– 7	14– 6	15– 6	15– 6
Aft.....	20	18– 6	18– 2	17– 6
Displacement ...	3,180	2,855	3,420	3,330

* Average of all ships.
† Crossed 180 degrees meridian.
‡ Computed from average of patent logs. (Course changed.)

Coal, Pocahontas from Government pile, Yokohama.
Ships' bottom clean.
Sea, moderate, 11 days; smooth, 3 days.

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upon reserve feed stored in double-bottom compartments. Main air pumps of the *Cincinnati* and *Raleigh* are independent, while those on the *New Orleans* and *Albany* are attached to and worked by beams from the main engines; an exhaust feed-water heater is fitted on the *Cincinnati* and *Raleigh*, but there is no such device on the other two ships.

From an inspection of the table it will be observed that during the first ten days of the run there was but little variation in speed, the average being 9.97 knots per hour. The following is a recapitulation of the important data during that time:

	Cincinnati.	*Raleigh.*	*New Orleans.*	*Albany.*
Average revolutions	72.1	76.7	75.1	77.8
steam pressure, boiler (gauge)	193	167.3	127.7	132.3
steam pressure, engine (gauge)	188	162.3	125.7	127.3
steam pressure, 1st receiver (absolute)	34.1	33.8	21.7	22.3
steam pressure, 2d receiver (absolute)	14.3	17.4	13.3	8.9
Vacuum	22.2	21.9	22.7	23.3
Temperature of feed (Fahrenheit)	199.2	180	111.5	124.1
Water distilled, per day, gallons	6,662	*...	3,055	2,400
I.H.P., (main engines only)	897	867	801	808
Coal per hour, pounds	2,831	2,853	3,063	3,288
per I.H.P. (main engines only), lbs	3.16	3.29	3.82	4.07
square foot grate, per hour, pounds	11.19	11.28	12.93	13.87

* Not recorded.

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BALANCE CYLINDERS.

By LUTHER D. LOVEKIN, ASSOCIATE.

[The following additional information on the subject of Balance Cylinders is in continuation of the article which appeared in Volume XVI, page 896, of the JOURNAL.]

For the benefit of those who may become interested in the balancing of valve gears I have taken the liberty of appending a recent extract from the "Marine Engineer," of London, England, under date of April 1, 1905, which is in answer to an article on "Slide Valve Resistances," by Mr. J. E. Cooper, in this same paper, under date of May 1, 1904, describing the Assistant Cylinders as fitted on the Royal Mail Steamer *Ophir*. These were Joy's Patent Assistant Cylinders, and give the reader a good opportunity for comparing the same with "The Improved Lovekin-Thom Assistant Cylinders," described previously, and as fitted on the Pacific Mail Steamer *Manchuria*. In order to determine the value of the Lovekin-Thom Assistant Cylinders under varying revolutions, I had several indicator cards taken while on the trial of the *Manchuria*, and made a direct comparison with the ordinary balance cylinder commonly employed under these same conditions. All cards have been plotted fully, and show at a glance *the great superiority of a properly designed assistant cylinder over the old form of balance cylinder, under all possible conditions*, such as varying revolutions, varying receiver pressure, etc., etc.

It will be noticed that with the use of the ordinary balance cylinder the *maximum pressure per square inch on all parts of the valve gear never falls below 51 pounds*, while with the use of the Improved Lovekin-Thom Cylinder we have as low as *11.3 pounds pressure per square inch*, or, approximately, only one-fourth that of the ordinary balance cylinder. Thus

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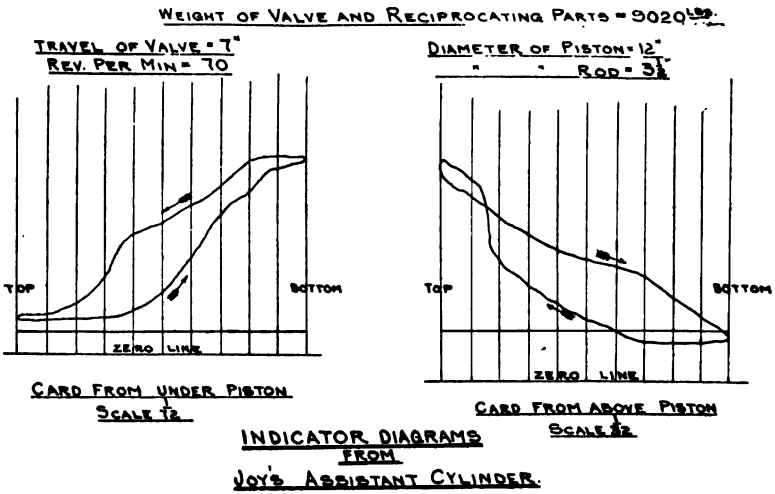
it will be seen that there is *everything to gain and nothing to lose by its general adoption on all reciprocating valve gears.*

Further attention is directed to the relative size and weight of these two cylinders, which are clearly shown on this same sheet.

SLIDE VALVE RESISTANCE.

To the Editor of the "Marine Engineer:"

Dear Sir: I thank you for the data supplied in response to my request published in your January issue. I desire also to thank Mr. Jasper E. Cooper for his courtesy in sending me



INDICATOR DIAGRAMS FROM JOY'S ASSISTANT CYLINDER.

FIG. I.

the indicator cards of the Joy Assistant Cylinder, which correspond with the data mentioned.

I append herewith the cards and the diagrams which I have plotted therefrom. I will explain all the steps thoroughly, so that they may be easily followed and verified:

First, divide the cards into a convenient number of parts, say ten, as would be done in the case of any indicator card to be worked up by hand. Draw a line below the atmospheric line to indicate absolute zero, and from this measure the ab-

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solute pressures at each point in the stroke, both up and down, as given by the card. Next multiply the pressures thus obtained by the area of the piston on which they act. Thus, the pressures measured from the card taken from the top of the piston will be multiplied by 113.1 square inches, this being the area of a piston 12 inches in diameter. For the bottom card multiply by 103.48 square inches, as this is the effective area after deducting the area of the rod.

We now have the absolute total pressures of steam acting above and below the piston at all points of the stroke. These are the forces which the steam sets up, one opposing the other. There is another known force, that of gravity, which acts constantly downward with a force equal to the weight of the valve and gear. In this case the weight is 9,020 pounds. It is evident that this force acts with the steam pressure on top of the piston, so these two must be added together to give the total downward force. The upward force consists solely of the pressure of the steam below the piston.

Proceeding now to lay out these forces in graphic form, we refer them to a common base line, as shown in Fig. 2. The pressure below the piston is plotted direct from each corresponding point in the indicator card. Of course, this is from the card from below the piston, and care must be taken to distinguish between the up and down strokes.

In order to plot the downward pressures, first take that due to gravity, or 9,020 pounds, and lay that off on a parallel line the proper distance above the base line. This represents the constant downward pressure due to the weight of the gear. In order to get the total downward pressure we must add that due to the steam on top of the piston. This is done in exactly the same manner as for the bottom steam pressure, except that it begins at the gravity line instead of at the base line.

We now have the upward and downward forces laid down so that they may be analyzed and compared. A little study of the indicator cards will show that the lines correspond properly, and any point in question can be verified by working up the data given.

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The curve on the diagram marked "Resultant" is obtained from the two curves of upward and downward forces by taking their differences at each ordinate, and measuring off that distance from the base line. Thus it will be noticed that in general the total downward forces exceed the upward forces. This is brought out clearly in the "Resultant" curve, which shows that only for a small portion of the stroke near the bottom does the line cross the base line, and indicate a resultant upward force on the eccentric strap. This resultant curve is easily verified by stepping off the vertical distance between the curves of upward and downward forces, and measuring that distance up or down from the base line according as the upward or downward forces predominate.

Thus far the analysis has been very simple, merely a little arithmetic and common-sense interpretation of the indicator diagrams. We must now introduce the force of inertia, which is not generally so well understood. Yet I trust I may put it so that all may understand its action in this case. We know that a stationary body requires force to overcome its inertia and start it moving, and a moving body requires force to overcome its inertia and stop it. This is represented in the diagram by the diagonal line crossing the base line at the middle of the stroke. At 70 revolutions per minute it takes a force of over 4,000 pounds to start the valve and gear downward, and the same force to stop it at the end of the stroke, and start it up again. At the middle of the stroke the force of inertia becomes nothing, for here it changes from one direction to the other. It has been retarding the acceleration of the valve and gear up to this point, but now begins to resist the slowing down that occurs during the last half of the stroke. We leave the weight of the gear out of the discussion, after having taken care of it as previously shown by including it as a fixed portion of the downward forces.

In order to locate the curve or line which indicates the force of inertia at each point in the stroke we must have recourse to higher mathematics. The expression for the valve of the inertia force at any point of the stroke is

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$$F = \frac{r\pi^2 N^2 W}{900\ G}\left(\cos\theta + \frac{\cos 2\theta}{n}\right)$$

where r = radius of eccentricity, in feet,
N = revolutions per minute,
W = weight of valve and reciprocating parts, in pounds,
G = force of gravity, equals 32.16,
θ = angle made by the eccentric with the line of travel,
n = ratio of length of eccentric rod to radius of eccentric, equals $\frac{r}{l}$.

(Refer to Fig. 3.)

The proof of this expression is so long that I will not attempt it here, except to show that it is equivalent to the expression "$rw^2 \cos\theta \times$ Mass," which Mr. Cooper used in his article.

For all practical purposes we may assume that the ratio "n" of the eccentric rod to the eccentricity is so large that the fraction $\frac{\cos 2\theta}{n}$ becomes negligible. This leaves the expression $\frac{r\pi^2 N^2 W}{900\ G}\cos\theta$.

Now if instead of $2\pi N$, I take w, the angular velocity as taken by Mr. Cooper, and instead of N revolutions per minute I take 60 multiplied by "n" revolutions per second, and, instead of mass, its equivalent $\frac{W}{G}$, the fraction reduces to $r\,w^2 \cos\theta$ multiplied by mass. This is exactly the expression used by Mr. Cooper in his article. Now the maximum value of F, the force of inertia, is found when $\theta = 0$ degrees, or when the valve is at the end of its stroke. Cos θ then becomes unity and $F = \frac{r\,\pi^2\,N^2\,W}{900\ G}$ or $F = .0003408\ r\ N^2\ W$.

From the data furnished by Mr. Cooper with the cards

$r = 3\frac{1}{2}$ inches $= .292$ feet,
$N = 70$ revolutions per minute,
$W = 9{,}020$ pounds.

Hence $F = 4{,}394$ pounds.

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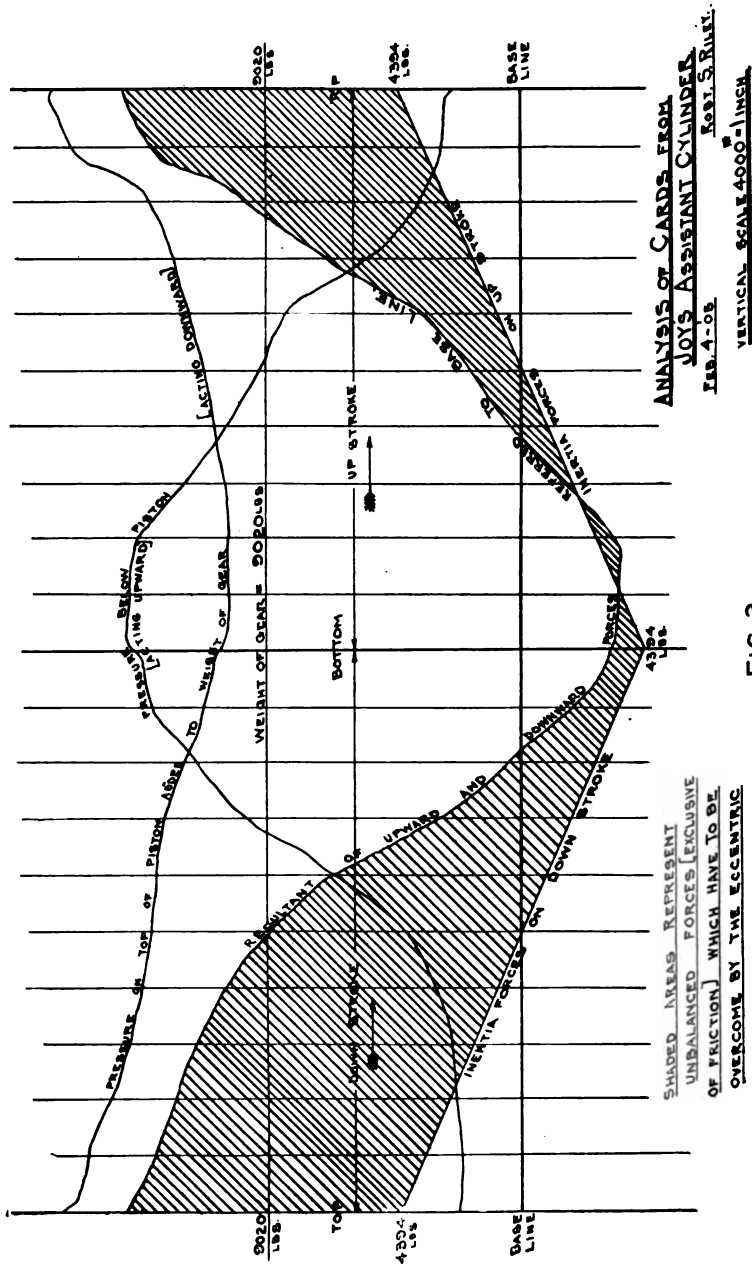
ANALYSIS OF CARDS FROM
JOY'S ASSISTANT CYLINDER
ROBT. S. RUST.
FEB. 4-08
VERTICAL SCALE 4000#-1 INCH
SHADED AREAS REPRESENT
UNBALANCED FORCES (EXCLUSIVE
OF FRICTION) WHICH HAVE TO BE
OVERCOME BY THE ECCENTRIC
9020 LBS.
4394 LBS.
BASE LINE
TOP
BOTTOM
UP STROKE
DOWN STROKE
WEIGHT OF GEAR = 9020 LBS
PRESSURE ON TOP OF PISTON ABOVE
PRESSURE (LIFTING UPWARD) BELOW PISTON
WEIGHT OF GEAR
ACTING DOWNWARD
RESULTANT FORCE UPWARD AND DOWNWARD
INERTIA FORCES ON DOWN STROKE
INERTIA FORCES ON UP STROKE
BASE LINE
FIG 2

Under the assumption that $\frac{l}{r}$ is infinitely large, the line indicating the inertia force is a straight line cutting the base line at its center and reaching the value of 4,394 pounds at each end of the stroke. We therefore lay off this line in the manner described in the diagram.

The ideal condition is attained when the resultant of gravity and the steam forces coincide with this inertia curve, for then all forces are exactly balanced.

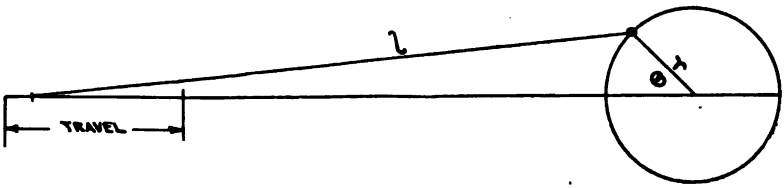

DIAGRAM TO ILLUSTRATE FORMULA FOR INERTIA FORCES

FIG. 3.

How far the Joy Assistant Cylinder fails to attain this ideal condition is shown by the shaded area between the two lines. Thus, on the first half of the down stroke and the last half of the up stroke there is an unbalanced downward pressure of nearly 10,000 pounds. This is more than the weight of the gear, showing that the assistant cylinder not only fails to relieve the stress on the eccentric, but actually adds to it.

I leave this without further comment to the consideration of your readers. I should be glad to hear from Mr. Cooper, or anyone else who can refute the facts shown up by this method of analysis.

Of course it is understood that I leave friction out of the discussion, just as Mr. Cooper did in his analysis. Could we obtain a satisfactory value for that force we might change the unbalanced forces materially, making them less on the down stroke and greater still on the up stroke.

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The fact is that the principle of admitting and exhausting steam, as done in the Joy Cylinder, is wrong. It often stops vibration, but that is because it imposes a more or less steady load as compared with no balance cylinder, or one of the ordinary type. The problem must be solved by a somewhat different action of the steam used. It can never be quite perfect, of course, although it is nearly so in a different type of assistant cylinder. I refer to the Lovekin-Thom improved type, as recently developed in America. But that is another story.

Thanking you, sir, for the space allowed me, and trusting that I may hear from anyone who fails to understand my analysis, I remain, yours truly,

ROBERT S. RILEY,
(*B. Sc., Extra First-Class Engineer.*)
236 West Logan Square, Philadelphia, Pa.
March 6, 1905.

THE LOVEKIN DIFFERENTIAL REGULATING VALVE FOR SUPPLYING STEAM TO A SERIES OF ASSISTANT CYLINDERS FROM THE H.P. RECEIVER.

It has been observed from actual practice that the receiver pressure in an engine varies approximately as the square of the revolutions. This is exactly the relation that is desired in the Lovekin-Thom Assistant Cylinders, viz: the pressure should vary as the inertia. Hence an ordinary reducing valve for keeping a constant pressure would be unsuitable except at one particular speed. On the other hand, the differential reducing valve is suitable at all speeds, for it will be seen upon examination that the pressure delivered by the valve bears a constant ratio to the receiver pressure from which the supply is taken. This ratio is fixed by the relative areas of the differential piston which operates the valve. It is much simpler than any other type of reducing valve; there is only one moving part, and nothing to get out of order.

The great advantage in using a valve of this type is that it enables us to reduce both the weight and the cost of the

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1- Used high p

2- These in

3- The st

4- At use

5- Wear a

6- No vi

7- No chan

8- Spring

9- Spring

10- Absol

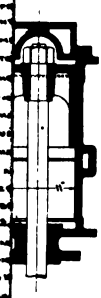
M ASSISTANT CYLINDER

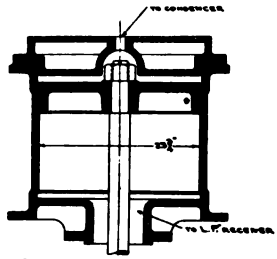
MANCHURIA" FOR L.P. VALVE GEAR

SCALE 1½"=1'0"

(WITHOUT VALVE STEM EXTENSION)

35 LBS.



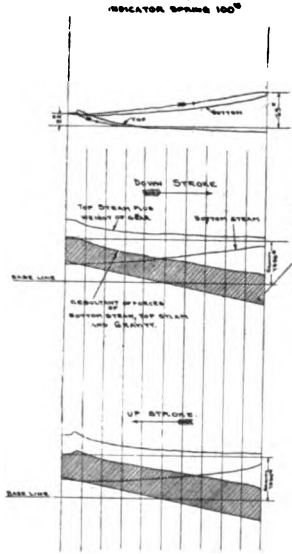


ORDINARY TYPE OF BALANCE CYLINDER
REDUCED FOR L.P. VALVE GEAR OF
"S.S. MANCHURIA"
SCALE 1½"=1'0"

TOTAL WEIGHT (WITHOUT VALVE STEM EXTENSION)

1740 LBS.

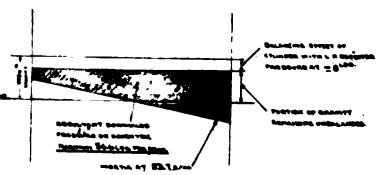
L.P. CUT OFF 29"
R.P.M. 53.7
INDICATOR SPRING 100#



MAXIMUM PRESSURE ON STEAM = 74.6 LBS PER SQ. IN.

EAM AT 6 LBS GUAGE = 18 LBS ABOVE

BOVE 10000 LBS = 1 INCH.



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assistant cylinders themselves where we have a series of cylinders of the same size and requiring the same pressure to each one. This is particularly noticeable in the cases of the cruiser *Washington* and the battleships *Kansas* and *New Hampshire* of the United States Navy. Each of these vessels have two four-cylinder triple-expansion engines, each engine having seven piston valves. The main parts of the valve gear, viz: eccentrics, eccentric straps, eccentric rods, links and blocks, valve rods and suspension rods, are interchangeable, and even the valves are of the same size on the intermediate- and low-pressure engines. Hence, we have practically the same inertia and gravity to overcome in each case, which means that all the assistant cylinders may be of the same diameter. It will be very apparent to all engineers that the cost of patterns will be greatly reduced under these conditions.

This regulating valve can be placed on the high-pressure receiver, which will necessarily give the highest and also the most definite pressure for maximum running conditions of the engine. Where there is only one cylinder to be supplied, such as in the case of the *Manchuria* low-pressure valve gear, all we need is a pipe led from the most suitable receiver direct to the assistant cylinder. This pressure is chosen while the diagram is being made, so as to give the best possible results, and in the case of the *Manchuria* it was found that the first intermediate receiver was the most suitable.

With the sleeve type of assistant cylinder as fitted on the Steamship *Texan* and also the second M.P. valve gear of the *Manchuria*, no pipes whatever were required. Each cylinder was supplied with steam from its own receiver, and was placed directly over the valve chest in a similar manner to the ordinary balance cylinder. It is very important to know that in no case can an assistant cylinder of proper design be placed on a valve gear, without having first worked out the necessary diagram for the particular case in hand, and no decision can be given as to which form of assistant cylinder is best suited to

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meet the ever-varying requirements, until such diagram has been accurately made and all the conditions fully considered.

In conclusion, I would state that no engine builder or designer would consider for a moment the running of a reciprocating engine without having first made *provision for proper compression at the top and bottom of the stroke.* Why, then, should not a valve gear receive its proper consideration *and thus save the engineer many worries at sea*, without mentioning the expense repairs, such as *relining the eccentric sheaves with white metal*, and the continual adjustment of the valve gear, all of which can be corrected by the proper use of these cylinders?

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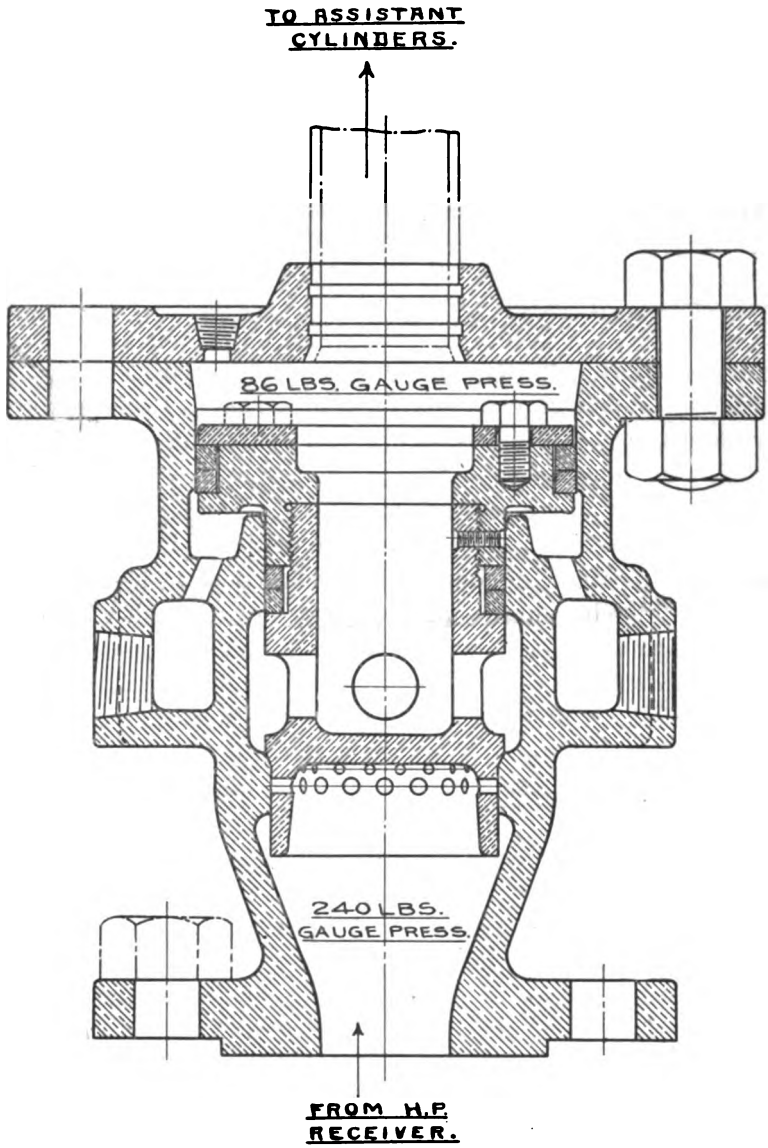

THE LOVEKIN DIFFERENTIAL REGULATING VALVE FOR SUPPLYING STEAM TO A SERIES OF ASSISTANT CYLINDERS FROM THE H.P. RECEIVER.

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THE VALUE OF THE FLAME OF COMBUSTIBLES.

By P. Mahler, Civil Engineer of Mines.

Translated from the "Revue Universelle des Mines," by B. F. Isherwood, Chief Engineer, U. S. Navy.

I.—VALUE OF FLAME COMBUSTIBLES.

The calorific power and the chemical compositions are, in general, sufficient elements of comparison between natural combustibles. These data permit, notably, the calculation of the value of flames, and the question may be asked whether this calculation has any practical utility.

The value of the flame of a combustible is the same thing as its temperature of combustion under constant pressure. It is measured by the thermometric degrees through which the gaseous products of the combustion are raised, supposing them to be heated by all the heat due to the combustion and solely by that heat. It must also be admitted that the combustion is complete and that it is realized by the aid of air under the atmospheric pressure, always nearly constant, as it takes place on grates.

The preceding definition is in accord as much as possible with industrial conditions; but, in fact, the temperature of combustion is a theoretical number, defining with precision a limit that practical industry cannot exceed with any given combustible and for the given bodies to be heated, and it seems interesting to compare different coals from this point of view.

The calorimetrical bomb and the elementary analysis having supplied* me with the necessary data for the calculation of temperatures of combustion, I have determined the value of the flames of a complete series of combustibles.

* "Etude sur les Combustibles," by P. Mahler, 1903, published by Beranger.

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II.—DETERMINATION OF THE TEMPERATURE OF COMBUSTION.

The determination of the temperature of combustion is done without difficulty, and the method I have followed is an old one. Nevertheless, I will cite the principal points, because they can be of service to engineers in many cases, as in the examination of the functioning of furnaces,* and of the economic results of gas motors calculated from the temperature of detonation of an explosive, etc.

Be q the calorific power of unit of weight of a combustible under constant pressure. Be C_m the mean specific heat† between O and T, absolute temperature of any one of the gases whatever composing the gases of combustion; let it be taken in relation to the weight of the gas which occupies a volume equal to 22.3 liters. Be N the number of molecules (22.3 liters) of this gas resulting from the combustion. As the energy of a gaseous mass depends wholly on its temperature, and as the coal in question is burned at a temperature very near 0° C. or T_0, the heating due to the combustion between T^0 and T_1 is, according to our hypothesis, united closely to the calorific power by the expression

$$q = \Sigma N(C_m T_1 - C_m T_0),$$

Σ signifying that the gases of combustion are a mixture of several gases for which N and C_m have special values.

The analysis of the combustible gives N, and if C_m were constant, we could easily deduce from the above equation T_1, since the temperature sought for $\theta = T_1 - 273°$ in Centrigrade degrees.

But the problem is more complex. Messieurs Millard and

* "Le Chauffage Industriel," by Mr. E. Damour.

† dq being the elementary heating of 1 kilogramme, the true specific heat at the temperature t is

$$\frac{dq}{dt} = c$$

by definition.

Between the true heat and the mean heat from o to t, we have, in general, the relation:

$$S_o^t c dt = C_m t, \quad c = \frac{dC_m t}{dt}.$$

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Le Chatelier have demonstrated that the specific heat of the gases of combustion increase with their temperature, this phenomenon becoming very sensible at high temperatures. According to these physicists it should be represented by a function

$$C_m = \alpha + \beta T.$$

The calculation of T_1 and, consequently, of θ, depends then on an equation of the second degree:

$$q = \Sigma N\,[(\alpha + \beta T_1)\,T_1 - (\alpha + \beta T_0)\,T_0]$$
$$q = \Sigma N\,[\alpha\,(T_1 - T_0) + \beta\,(T_1^2 - T_0^2)]. \qquad (1)$$

Messieurs Mallard and Le Chatelier have deduced from their experiments the values of α and of β relative to the different gases.*

I ought to say that in these later days certain very exclusive partisans of the law of Clausius regarding specific heats have contested the law of increase shown by Messieurs Berthelot and Vieille and by Messieurs Mallard and Le Chatelier. The law of increase, nevertheless, does not for this reason appear any less conformable to the truth. And if we reflect on the considerable difference existing between the temperatures of combustion calculated according to this law and the highest temperatures obtained in the best industrial furnaces (Martin-Siemens' furnace, 1,600°), we may even suppose—for the effect of the dissociation is feeble—that the increases resulting from the known experiments are inferior to the truth.

Under any circumstances the exactness of the experiments of Messieurs Mallard and Le Chatelier certainly surpass the precision I need for the comparative investigation that is the subject of this note.

Which being premised, we will now return to the calcula-

* Researches, experimental and theoretical, on the combustion of explosive gaseous mixtures; third memoir on the temperatures of combustion and the specific heats of gases at high temperatures, by E. Mallard and H. Le Chatelier ("Annales des Mines," 1885).

Report on the study of the questions relative to the use of explosives in presence of grisou (fire damp); temperature of detonation; by E. Mallard ("Annales des Mines," 1888).

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tion of the temperature of combustion of coals and of other natural combustibles.

The gases of combustion comprise in general only perfect gases such as azote, and incidentally the oxide of carbon, and the condensed gases steam and carbonic acid. Relatively to the perfect gases, the remark must be made that their specific heats are nearly equal at all temperatures between the ordinary atmospheric temperatures and the temperature 3,000°.

According to the above physicists the following are the coefficients for the above gases: *

$$\text{Perfect gases} \begin{cases} \alpha = 6.5 \times 1{,}000^{-1} \\ \beta = 0.6 \times 1{,}000^{-2} \end{cases}$$

$$\text{Superheated steam} \begin{cases} \alpha = 6.5 \times 1{,}000^{-1} \\ \beta = 2.9 \times 1{,}000^{-2} \end{cases}$$

$$\text{Carbonic acid} \begin{cases} \alpha = 6.5 \times 1{,}000^{-1} \\ \beta = 3.7 \times 1{,}000^{-2} \end{cases}$$

From these result the following expressions of the heating of a molecule of the above gases from T_0 to T_1 in large calories.†

* "Cours de Chimie Industrielle," by H. Le Chatelier.

† There may be needed the expression of the heating of a constant volume; for example, for the calculation of the temperatures of detonation, and also for the study of gas motors.

In this case, the coefficients of $[T_1^2 - T_0^2]$ are the same as above, and the coefficient of $[T_1 - T_0]$ is solely

$$4.5 \frac{T_1 - T_0}{100}.$$

There may be also need of the heating of a molecule of methane, as follows:

$$4.5 \frac{T_1 - T_0}{1{,}000} + 6 \frac{T_1^2 - T_0^2}{1{,}000}$$

at constant volume.

The difference between the heating at constant volume results from the difference between the two specific heats $[Cp, Cv]$ a difference due to the quantity of heat absorbed by the work of dilation accompanying the increase of temperature.

We know, in fact, that the two heats of the heating are connected by the expression:

$$q = Q + A \int_{v_0}^{v_1} p_0 \, dv, \text{ or } q - Q = A \int_{v_0}^{v_1} p_0 dv.$$

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The values of the heating of one molecule of gas and the knowledge of the numbers N of molecules disengaged by the combustion, enables us to translate into figures the equation (2). Further, this equation will give directly the heat of combustion sought for in degrees centigrade; if we put

$$T_0 = 273; \; T_1 = 0 + 273,$$

it becomes then

$$q = \Sigma N \left[\frac{a\theta}{1{,}000} + \beta \frac{(\theta + 273)^2 - 273^2}{1{,}000^2} \right]$$

in large calories, and we can solve it algebraically. But when we have to determine a great number of temperatures, as in this investigation, it is preferable to employ the following artifice:

Briefly, it consists in finding a positive value of θ, so that the number of calories expressed by Σ equals the number of calories q of the calorific power.

Arrange then the table of the heating of one gaseous molecule from 0° to 100, 200° C., etc., from 100 to 100° for example. Extend the table to the perfect gases, to superheated steam, and to carbonic acid. This operation is easy by means of the

---

Taking account of the equation $pv = RT$ and the definitions of Q and of q, there exists between the two specific heats at constant pressure and constant volume.

$$(Cp - Cv)\,(T_1 - T_0) = Ap_0\,(v_1 - v_0) = \frac{Ap_0 v_0}{273}\,(T_1 - T_0),$$

or between the molecular heats:

$$Cp - Cv = \frac{Ap_0 v^0}{273}.$$

Taking as units the kilogram and the decimeter:

$$A = \frac{1}{E} = \frac{1}{4{,}260}, \quad p = 103.33, \quad v_0 = 22{,}300, \quad Cp - Cv = 1.98,$$

about two calories.

Perfect gases, $q = 6.5 \dfrac{T_1 - T_0}{1{,}000} + 0.6 \dfrac{T_1^2 - T_0^2}{1{,}000^2}$

Superheated steam, $q = 6.5 \dfrac{T_1 - T_0}{1{,}000} + 2.9 \dfrac{T_1^2 - T_0^2}{1{,}000^2}$

Carbonic acid, $q = 6.5 \dfrac{T_1 - T_0}{1{,}000} + 3.7 \dfrac{T_1^2 - T_0^2}{1{,}000^2}$

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formulas we have given. Further, the calculation can be very rapidly made; the squares

$$[T + 100]^2,\ [T + 200°]^2,\ [T + 300°],$$

etc., are immediately deduced, one from the other.

The numbers N, on the other part, being known, it is very evident that the table in question enables the value of the total heating of the residue of combustion (azote, carbonic acid, superheated steam), corresponding to 100, 200° C., etc.

Also, as we have always an idea approximately of the temperature sought for, it suffices to calculate the value of Σ for temperatures near θ, in order to determine the quantities of heat q_1 and q_2, such as

$$q_1 < q < q_2$$
$$\theta_1 < \theta < \theta_1 + 100.$$

The temperature sought for is comprised between θ and $\theta_1 + 100$, it differs from θ_1 by a quantity E, $\theta = \theta_1 + E$. Now E is easily calculated with sufficient precision.

In fact, betwen θ and θ_1, + 100 there can be admitted that the increase of the temperature is proportional to the increase in the quantity of heat, and that, in this interval, 1 calorie corresponds to an augmentation of temperature:

$$\frac{100}{q_2 - q_1}.$$

E, which corresponds to an increase of heat $q - q_1$, is, under these conditions, represented by

$$(q - q_1)\frac{100}{q_2 - q_1},$$

and the approximate value of θ is

$$\theta = \theta_1 + (q - q_1)\frac{100}{q_2 - q_1}.$$

The solution is not rigorous, but a greater exactness is useless.

A further reproach to this method of determination is that it does not take into account the dissociation of the superheated steam and of the carbonic acid. Regarding

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this objection, however, there may be stated that the dissociation of the superheated steam between 0° and 2,500° has a negligible influence. The effect of the dissociation of the carbonic acid is, also, practically but little sensible up to 2,000°, a temperature near which are grouped the most part of the numbers that I give later on. Two of these numbers are near 2,500°, and it is proper to say that the coefficients of dissociation of the carbonic acid seem to have a real importance in the neighborhood of that temperature.*

II.—TABLE OF THE HEATINGS IN CALORIES.

The table of the heatings of the molecular volume of the gases is a useful document for many purposes. I think it

Heatings in calories of the molecular volume of gases from 0 degree centigrade to 3,000 degrees centigrade (under constant pressure.)

The molecular volume corresponds to 22.32 liters. The molecules are represented by the symbols Az^2, O^2, H^2, CO, CO^2.

Temperatures, degrees centigrade.	Azote, oxygen, oxide of carbon, hydrogen.	Superheated steam.	Carbonic acid.	Temperatures, degrees centigrade.	Azote, oxygen, oxide of carbon, hydrogen.	Superheated steam.	Carbonic acid.
0	0.00	0.00	0.00	1,600	12.46	20.35	23.09
100	0.68	0.83	0.87	1,700	13.34	22.13	25.18
200	1.39	1.73	1.85	1,800	14.23	23.93	27.31
300	2.10	2.67	2.87	1,900	15.14	25.83	29 55
400	2.82	3.69	3.99	2,000	16.05	27.76	31.83
500	3.56	4.76	5.17	2,100	16.98	29 74	34.18
600	4.31	5.89	6.44	2,200	17.92	31.81	36 64
700	5.07	7.07	7.77	2,300	18.87	33.91	39.14
800	5.85	8.30	9.16	2,400	19.84	36.10	41.75
900	6.63	9.62	10.66	2,500	20.81	38.32	44.40
1,000	7.43	10.98	12.12	2,600	21.80	40.62	47.16
1,100	8.24	12.40	13.85	2,700	22 80	42.95	49 96
1,200	9.05	13.87	15.55	2,800	23 82	45.37	52.87
1,300	9.89	15.41	17.33	2,900	24.84	47.82	55.81
1,400	10.73	17.00	19.18	3,000	25 88	50.35	58.86
1,500	11.59	18 65	21.11				

* The coefficient of dissociation is the proportion of the gas dissociated to that which would exist if the decomposition were complete. For carbonic acid, and at 2,500°, these coefficients are approximately 0.2 (pressure of 1 atmosphere), 0.4 (pressure of $\frac{1}{10}$ of an atmosphere), etc.

See Experimental Researches on Chemical Equilibriums, by Le Chatelier, "Annales des Mines," 1888.

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beneficial to publish the one which I have used; its figures have been carefully collated, and it is more extensive and nearer complete than the most part of those heretofore published.

III.—EXAMPLES OF DETERMINATION OF HEATS OF COMBUSTION.

Our hypotheses suppose that the combustibles are burned under the normal atmospheric pressure and with the aid of air, at the temperature of 0° C.

One hundred volumes of air decompose into about 80 volumes of azote and 20 volumes of oxygen. Consequently, if a molecular volume of oxygen, O^2, is necessary for the burning of a certain combustible, the gas resulting from the operation will comprise, specially, 4 molecular volumes of azote or $4Az^2$.

In order to commence, take the simplest case, that of the combustion of hydrogen.

(1) *Temperature of Combustion of Hydrogen.*—According to the above conventions, the combustion of hydrogen in cold air will be expressed as follows:

$$H^2 + \frac{O^2}{2} + 2Az^2 = 2Az^2 + H^2O, \text{ gas}, + 58.2 \text{ calories.}$$

The residue of the combustion of 1 molecule (or 2 grams) of hydrogen comprises, then, 2 molecules of azote and 1 molecule of vaporized water. The heat disengaged by the reaction is 58.2 calories $= q$.

By referring this to the table of molecular heatings, there are easily obtained the values of Σ at about the anticipated temperature θ of combustion.

Heating at (degrees),	1,800	1,900	2,000
Of 2 molecules of azote,	$2 \times 14.23 = 28.46$	$2 \times 15.14 = 30.28$	$2 \times 16.05 = 32.10$
Of 1 molecule of steam,	$= 23.93$	25.83	27.76
	52.39	$q_1 = 56.11$	$q_2 = 59.86$

The temperature sought for is, then, between 1,900° and 2,000° C., corresponding respectively to the heatings in calories 56.11 and 59.86. Finally, we have, by applying the

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formula established above and taking account that $q = 58.2$ calories,

$$\theta = 1{,}900 + (q - q_1)\frac{100}{q_2 - q_1} = 1{,}900 + 2.09\frac{100}{3.75} = 1{,}956^\circ \text{ C.},$$

or, in round numbers, 1,960° C.

We should find for the oxide of carbon, very approximately, 2,100° C., the flame of oxide of carbon is, then, hotter than the flame of hydrogen.

(2) *Temperature of the Combustion of a Peat.*—Analysis of 100 grams of the specimen remaining after abstraction of ash and hygroscopic water:

Carbon	57.21
Hydrogen	5.69
Oxygen and azote	37.10
	100.00

Calorific power, water vaporized, 5,590 calories, corresponding for 100 grams to 559 calories.

The gases resulting from the combustion of 100 grams, in air, of this peat comprise carbonic acid, water vaporized, and azote, of which the above analysis permits the determination of the number N of molecular volumes.

The proportion of carbonic acid is deduced from its tenor in carbon, as we know that 12 grams of carbon give, in burning, a molecular volume of carbonic acid, there will be in the gases of combustion

$$\frac{57.21}{12} = 4.76$$

volumes of carbonic acid.

Similarly, since 2 grams of hydrogen produce a molecule of water, the proportion of water vaporized will be

$$\frac{5.69}{2} = 2.84$$

volumes.

To calculate the azote, it is necessary to proportion it to the quantity of oxygen strictly utilized for the combustion.

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	Volumes of oxygen.
Now, with 4.76 volumes of carbonic acid, there are combined	4.76
And, with 2.84 volumes of water vaporized, there are combined	1.42
Total	6.18

Deducting from this sum the corresponding volume of oxygen pre-existing in the peat

$$\left(\frac{37.10}{32} = \right) 1.16,$$

and neglecting the small quantity of azote which accompanies this body in the peat, it becomes 6.18 — 1.16 = 5.02, or about 5 volumes. Five molecular volumes of oxygen are then necessary to burn the peat in question; these 5 volumes correspond to 25 volumes of air, containing 20 volumes of azote which pass unchanged into the gases of combustion.

To sum up: The residue of combustion will have the following composition in molecular volumes:

	Volumes.
Azote	20.00
Vaporized water	2.84
Carbonic acid	4.76
	27.60

Possessing thus the numbers N of gaseous molecules, we determine in the following manner the values of Σ in the neigborhood of the temperature of combustion sought for:

Heatings at (degrees)—	1,900.	2,000.	2,100.	2,500.	2,600.
Of 20.00 volumes of azote	302.80	321.00	339.60	416.20	436.00
Of 2.84 volumes of steam	73.35	78.83	84.46	108.80	115.40
Of 4.76 volumes of carbonic acid	140.65	151.51	162.69	211.34	224.50
	516.80	$q_1 = 551.34$	$q_2 = 586.75$	736.34	775.90

As on the other part, $q = 559$, the temperature of combustion sought for is visibly placed between 2,000° and 2,100°, we have, finally, with very close approximation,

$$\theta = 2{,}000 + (559 - 551)\frac{100}{587 - 551} = 2{,}020^\circ \text{ C.}$$

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The values above of the heatings at 2,500° and at 2,600° enable an idea to be had of the influence that would be exerted on the value of the flame by the previous heating of the air to 1,000° for example. In this hypothesis, the quantity q would be formed not only of the number of calories due to the heat of the combustion, but also of the number of calories carried by 25 volumes of air heated to 1,000°, $q = 559 + 25 \times 7.43$; according to the table of the heatings $q = 559 + 186 = 745$ calories. Completing the calculation, we find $\theta = 2{,}520°$. The previous heating of the air to 1,000° has raised the temperature of combustion about 500° C.

(3) *Temperature of Combustion of a Coal.*—Take, for example, the coal of Treuil (St. Etienne).

The elementary analysis of this coal has given abstraction made of ash and hygroscopic water:*

Supposing the water resulting from the combustion to be vaporized, the calorific power is 8,580 calories.

The method of calculation employed for the peat is here applicable without modification. We find, notably in the gaseous residue, the following composition in molecular volumes:

Azote	34.00
Superheated steam	2.50
Carbonic acid	7.40
Total	43.90
corresponding to 42.50 volumes of air used in the combustion.	
Value of the flame (cold air), degrees	2,010
Value of the flame (air at 1,000 degrees), degrees	2,570

I add that the reader will find other examples, notably in what concerns gaseous combustibles, in Mr. Damour's book, already cited.

* Combustibles, such as coal and peat, always contain a certain quantity of hygroscopic water which passes entirely into the gases of combustion. It is very easy to take account of this water in the calculation of the temperature.

Carbon	89.23
Hydrogen	5.03
Oxygen and azote	5.74

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IV.—RESULTS OF CALCULATIONS.

I have applied the preceding method to the calculation of the value of the flame of a certain number of combustibles. In order to render more easy the comparison of the results, I have supposed these combustibles to be deprived of their ash and hygroscopic water.

I have taken the elementary analyses and the calorific powers from pages 54, 59 and 80 of my work, entitled "Etudes sur les Combustibles." The following table gives the figures obtained.

The value of the flame varies very little from one combustible to another. The mean of sixteen enumerated is 1,986°, and it seems that the round number 2,000° can be adopted as representing the theoretical temperature of combustion of any combustible whatever of vegetable origin. As absolutely rigorous constants could not be obtained for the process I have described, a more precise number is useless.

Designation of the combustibles.	Calorific power.		Value of the flame.
	Water condensed.	Water vaporized.	
	Calories.	*Calories.*	*Degrees, C.*
Oak from Lorraine	4,690	4,370	1,865
Peat from Bohemia	5,900	5,590	2,020
Lignite from Trifail	6,650	6,370	1,960
Flaming coal from Blanzy (Ste. Marie)	8,350	8,060	1,990
Flaming coal from Decazeville (Tramont)	7,840	7,530	1,960
Oxidized coal from Commentry	6,380	6,200	1,960
Gas coal from Commentry	8,410	8,110	1,950
Gas coal from Béthune	8,670	8,380	1,990
Gas coal from Lens	8,740	8,450	2,010
Fat coal from St. Etienne (Treuil)	8,860	8,580	2,010
Marechale coal from Roche-la-Moleire	8,860	8,600	2,030
Semi-fat coal from d'Anzin (St. Marc)	8,660	8,430	1,980
Anthracite coal from Commentry	8,460	8,290	2,030
Anthracite coal from Kebao	8,530	8,370	2,020
Anthracite coal from Creusot	8,690	8,480	2,010
Anthracite coal from Pennsylvania	8,266	8,140	2,000

The value which, in the above table, is the farthest from the mean of 2,000°, is that of oak—1,865°.

The temperatures which are the farthest removed in excess from the mean 2,000° are, in general, those of the anthracite

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coal. The popular opinion has always been that anthracite has a hotter flame than coal. The relative value of the flames sustains this opinion.

To the results in the above table can be added those which I have found for different liquid combustibles, as follows:

Ethylic alcohol and methylic alcohol, degrees	1,700
Amylic alcohol, degrees	1,850
Crude American petroleum	2,000 degrees, (calorific power, 10,400.)
Essence of American petroleum	1,910 degrees, (calorific power, 10,270.)
Refined American petroleum oil	1,660 degrees, (calorific power, 10,280.)

Finally, below will be found the temperatures of combustion of several gases under constant pressure:

Hydrogen, degrees	1,960
Oxide of carbon, degrees	2,100
Methane, degrees	1,850
Acetylene, degrees	2,350
Illuminating gas, mean*, degrees	1,950
Water gas, industrial, degrees	2,000

An examination of the numbers grouped in this work shows the apparently paradoxical appearance that of two combustibles, the one having the greatest calorific power is not necessarily the one having the hottest flame.

The flame of the essence of petroleum is relatively cold. The flame of the sample of peat appears hotter than that of the coal of the Treuil, the calorific power of which is excellent. The flame of the oxidized coal from Commentry is a little better than that of the same coal not deteriorated.

Moreover, everything occurs in all cases, and, notably, for the combustibles of vegetable origin, as if the elements of the temperature of combustion, calorific power, elementary composition, volume and composition of the gases of combustion, compensated each other so as to give to all the combustibles flames of very nearly the same thermometric degrees of temperature.

This, however, does not lessen the advantages due to the high calorific power of good coals, and which is the true

*According to Mr. A. Sainte-Claire Deville (Société Technique de l'Industrie du Gaz," 1903.)

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measure of their value, since the flame utilizable of a coal is a great deal more voluminous than that of a peat, a very much oxidized matter, the combustion of which requires but very little air.

We have seen, for example, that the combustion of 100 grams of Bohemian peat corresponds to the disengagement of 28 volumes of gases of combustion at 2,000°, while the gaseous residue of the coal from the Treuil attains 44 units* at the same temperature.

* The use of hot air for improving the flame, from the thermometric point of view, is certainly more efficacious in the furnaces in which coal is burned than in those in which an inferior combustible is burned.

—Reprinted from "Journal of the Franklin Institute."

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TEST OF THE MACHINERY OF THE *JAMES C. WALLACE* UNDER SUPERHEATED AND SATURATED STEAM.

By Commander W. W. White, U. S. N. (Retired), Member.

The economical advantages in using superheated steam have been recognized for many years, but not until recently has steam engine and steam turbine construction advanced to the point where, from a practical standpoint, the ill effects met with in past years have been overcome. Formerly, the impossibility of obtaining satisfactory metallic and other packing for rods and joints subjected to high temperatures, with the numerous attendant evils, as well as a proper cylinder lubricant, served to delay its general use. Of late years, however, such difficulties have largely disappeared, and in modern practice a temperature of 500 degrees Fahrenheit is readily and successfully handled in steam engines. In foreign countries, notably in Germany, a much higher degree of superheat is being used, but the limit, thus far, in the United States appears to be as stated. This, no doubt, results from the additional expense involved in up-keep incidental to high temperatures.

For the purpose of securing comparative data with superheated and saturated steam, used in the same engine, and by courtesy of the Acme Steamship Company on the Great Lakes, a Board was appointed, under instructions from the Bureau of Steam Engineering, to carry out tests of machinery on the steamer *James C. Wallace.* This vessel, which is one of the largest freighters on the Lakes, has lately been put into service. She is equipped with two Babcock & Wilcox marine water-tubular boilers with superheaters, and the arrangement is such that the latter may be dispensed with by a few simple changes, and saturated steam used.

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The Board was composed of Commander W. W. White, U. S. N. (Retired); Lieutenant Commander B. C. Bryan, U. S. N., and Lieutenant H. T. Winston, U. S. N.

A description of this vessel appeared in the JOURNAL, Vol. XVII, page 579. It will be only necessary, therefore, to repeat here the principal dimensions.

HULL.

The hull is of mild steel and was built by the American Shipbuilding Company, of Cleveland, Ohio. A cargo hold, hopper shaped, with girder arches, and without bulkheads, divisions or stanchions, constitutes an important departure and improvement in design. This hopper extends 408 feet in length, and measures 47 feet at the top and 29 feet at the bottom. The adoption of this construction makes it possible, by the use of automatic clams, to unload cargo without hand shoveling.

Length over all, feet	552
Beam, feet	56
Depth, feet	31
Number of cargo hatches	32
Capacity of water bottoms, tons	8,000
Coal bunker capacity, about, tons	350

MACHINERY.

The propelling machinery, as is usual in Lake vessels, is placed in the extreme after part of the vessel. The boilers are immediately forward of the engine, with an athwartship coal bunker forward of the boiler-room bulkhead.

ENGINE.

The main engine is of the quadruple-expansion, vertical, direct-acting, inverted, jet-condensing type. One piston valve, actuated by a Joy valve gear, is fitted on the side for each of the high, first and second intermediate-pressure cylinders. The low-pressure valve is a double-ported slide, placed on the after side of the cylinder and operated by eccentrics and a double-bar Stephenson link. All cylinders are unjacketed.

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Number of cylinders	4
Diameter of H.P., inches	18½
1st I.P., inches	28½
2d I.P., inches	43½
L.P., inches	66
Stroke (common), inches	42
Diameter of piston rods (one for each cylinder), inches	5⅝
Order of cylinders from forward: (1) H.P.; (2) 1st I.P.; (3) 2d I.P.; (4) L.P.	
Designed I.H.P	2,000
piston speed, feet	560

AUXILIARY MACHINERY.

The following shows the important dimensions of the auxiliary machinery in use during the tests, all of which is independent of the main engine. In addition, in Test II, two fire-room blowers (double simple engine 6 inches × 5 inches), and two small duplex pumps (2 inches × 1¼ inches × 1¾ inches stroke), which furnished water circulation through the blower bearings, were in operation.

Number.	Purpose.	Make.	Type.	Diameter of cylinders.			Stroke.
				H.P.	L.P.	Water.	
1	Main air pump.	Blake.	Vertical compound.	12	20	24	18
1	Main feed pump.	Blake.	Horizontal, compound, duplex plunger.	6	10	5	10
1	Water service pump.	Blake.	Horizontal duplex.	...	4½	3¾	4
1	Bilge pump.	Blake.	Horizontal duplex.	...	7½	8½	6
1	Stoker engine.	...	Vertical, one-cyl., simple.	...	6½	...	6

A feed-water heater is fitted in the engine room, into which all auxiliaries exhaust. The heater is cylindrical, with 2-inch tubes, through which the feed water is forced in its passage to the boilers. No attempt was made to determine the power of any of the auxiliaries in use.

BOILERS.

Steam is supplied by two Babcock & Wilcox boilers of the latest type, designed for a pressure of 250 pounds, with super-

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heaters and stokers. Plate I shows a sectional, and Plate II a front view of one of the boilers.

The superheater, it will be observed, consists of an upper and lower cross box having a number of bent tubes expanded into them. Steam, after being generated, is led from the drum by an outside pipe to the forward end of the superheater. After passing through the latter, it is directed through another outside pipe, at the after end, to the main steam pipe. One partition is placed in each cross box, thus causing the steam to flow three times from one box to the other in its passage from inlet to outlet.

Suitable valves are installed in the piping, so that the generated steam may be sent either through the superheater, or, if desired, directly to the main steam line.

Length of boiler proper, feet	11
including projection of stoker, feet	14
Width of boiler, feet and inches	12– 2
Height of boiler to top of drum, feet	15
superheater, feet and inches	15– 8
Diameter of drum, inches	42
Length over heads, feet and inches	13– 5
Thickness of shell plates, inch	$\frac{19}{32}$
Diameter of boiler tubes, inches	4
Thickness of boiler tubes	6 B. W. G.
Length of tubes between headers, feet	10
Number of sections wide, each boiler	18
including side sections	20
Diameter of superheater tubes, inches	2
Thickness of superheater tubes	10 B. W. G.
Square feet of grate surface, each boiler	74
heating surface, each boiler	2,900
superheating surface, each boiler	414
Ratio of G. S. to H. S.	1 to 39.19
including superheater	1 to 44.78

STOKERS.

Each boiler is provided with two chain stokers of the Crowe pattern, all driven by a single-cylinder engine 6½ inches × 6 inches (installed in the engine room and run at constant speed) through shafting, sprocket wheels, chain, eccentrics, &c. The speed of each stoker grate is independently controlled by an

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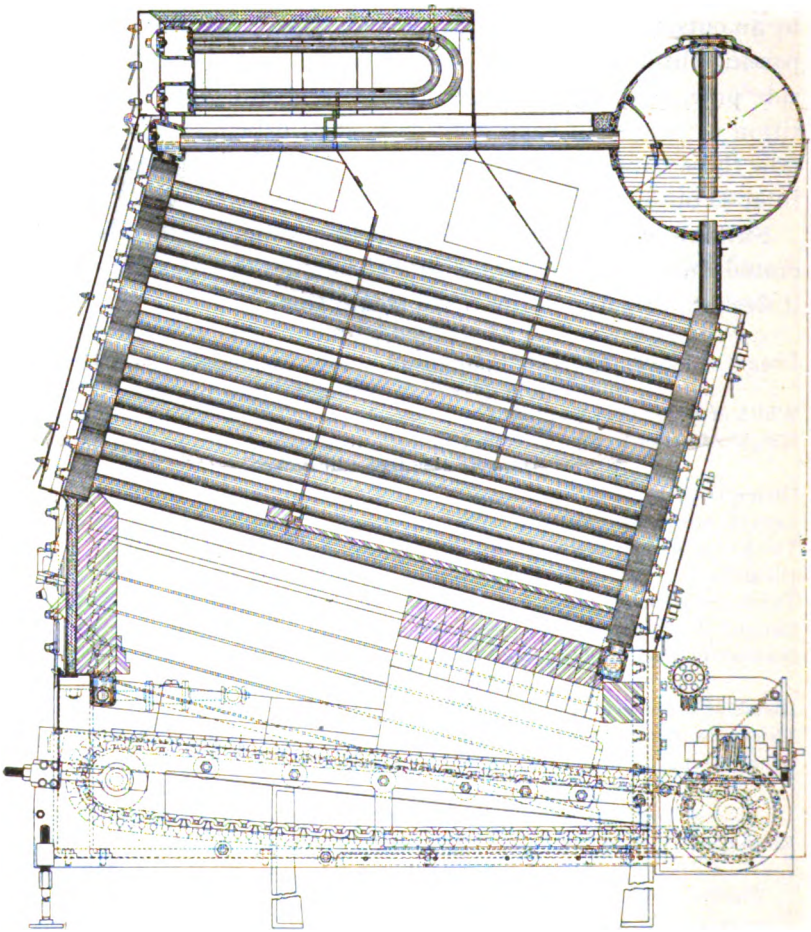

Plate I.—BABCOCK & WILCOX BOILERS, S. S. "JAMES C. WALLACE."

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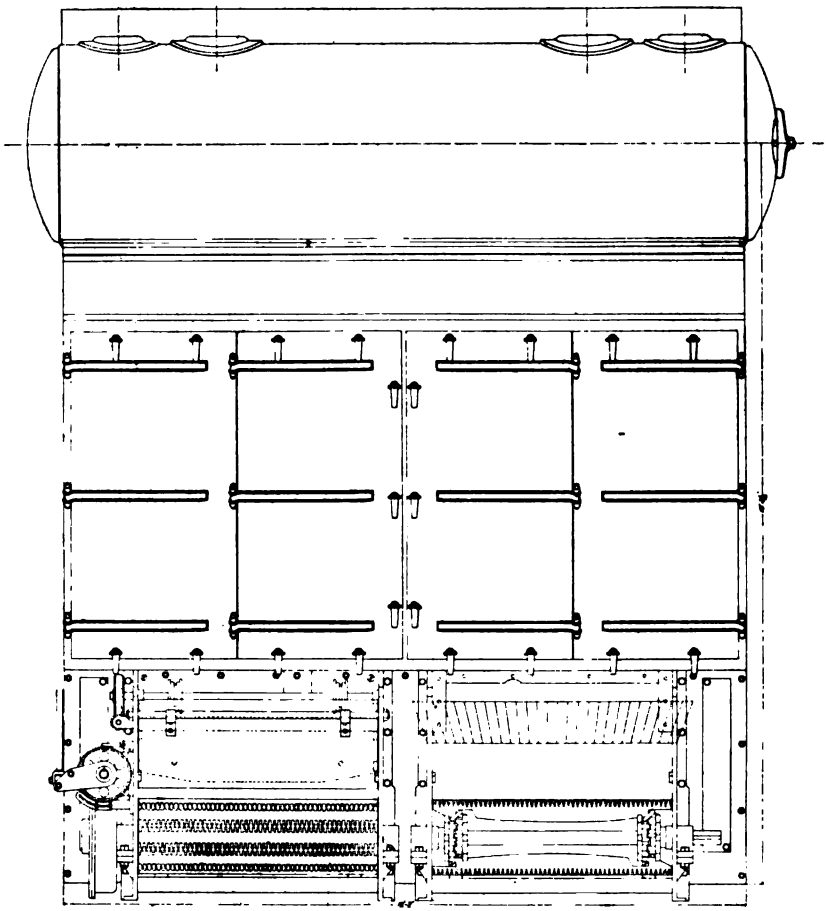

Plate II.—BABCOCK & WILCOX BOILERS, "S. S. "JAMES C. WALLACE."

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adjustable ratchet device, and its rate of travel is regulated, as frequently as circumstances require, in order to maintain steam at the proper pressure. Coal is fed into hoppers, at the front of the boilers, the thickness of the fire being governed by an adjustable plate extending across the width of the grate. The speed of the grate should be such that the coal, in passing from front to back, is entirely consumed. Ashes and clinkers are dumped and removed at the back of the boilers.

It may not be out of place to note here that the stokers worked perfectly during the entire trip, and so easily was the regulation of the coal supplied the furnaces controlled, that fluctuations in steam pressure were but slight indeed. No smoke issued from the smoke pipe at any time.

Where, as is the case on the *Wallace*, the steam supply demanded, and, consequently, the speed developed, is fixed, and only changeable within narrow limits, there can be no question as to the decided superiority of stokers over hand firing from every standpoint. Moreover, practical experience with stokers fitted to boilers of Lake steamers during the past few years has resulted in improving the mechanical construction to such an extent that breakdowns are rare, and when they do occur, are usually readily and quickly remedied. In view of this success, the advisability of equipping at least one-half the boiler plant of men-of-war with stokers is worthy of serious consideration; flexibility could be provided for by hand firing on the remaining boilers.

COAL.

The coal used during the entire trip was western Pennsylvania bituminous slack, loaded at Ashtabula. On each test an average sample was secured and sealed in a jar for determination, at a convenient time, of moisture, heat value, &c.

Three samples have been examined, and the heat value in British thermal units (by Mahler bomb calorimeter) per pound of dry coal, and proximate analyses are recorded below:

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	B.T.U.	Percentages of dry coal.		
		Volatile matter.	Fixed carbon.	Ash.
Sample No. 1, .	13,466	29.23	59.53	11.23
Sample No. 2, .	13,499	28.37	57.51	14.42
Sample No. 3, .	13,593	28.60	60.63	10.77
Average, . .	13,519	28.73	59.22	12.14

DESCRIPTION OF TESTS.

For comparison, two tests (tables I and II) were made with the superheater in operation, and two (tables III and IV) with saturated steam, the superheater being entirely blocked off and cut out. During all tests, the coal was carefully weighed, the main engine indicated, and the important engine and fire-room data recorded half hourly. The water evaporated was not weighed, as the facilities for doing so were lacking. This is to be regretted, since valuable information regarding steam economy of the main engine under the different conditions was not obtainable.

A blower, having a central inlet and discharging at its periphery, is fitted (see plates III and IV) at the back and over each boiler, to handle the products of combustion after passing through the boiler. In the ordinary steaming of the ship these blowers are used, as the draft is thus under complete control, and entirely independent of weather conditions. These blowers, however, are not an absolute necessity, and may or may not be used.

Plate III is an outline sketch of the boiler, in which is indicated the path of the products of combustion with the superheater in operation. As shown, a damper is provided in the uptake, which is kept closed or open, depending on whether the blowers are running or stopped. Of the two tests with superheated steam, the first was conducted with the blowers stopped, and the second with blowers (plate III) running. Both of these tests were made on the up trip of the vessel (Ashtabula, Ohio, to Duluth, Minn.) with no cargo, but with water ballast.

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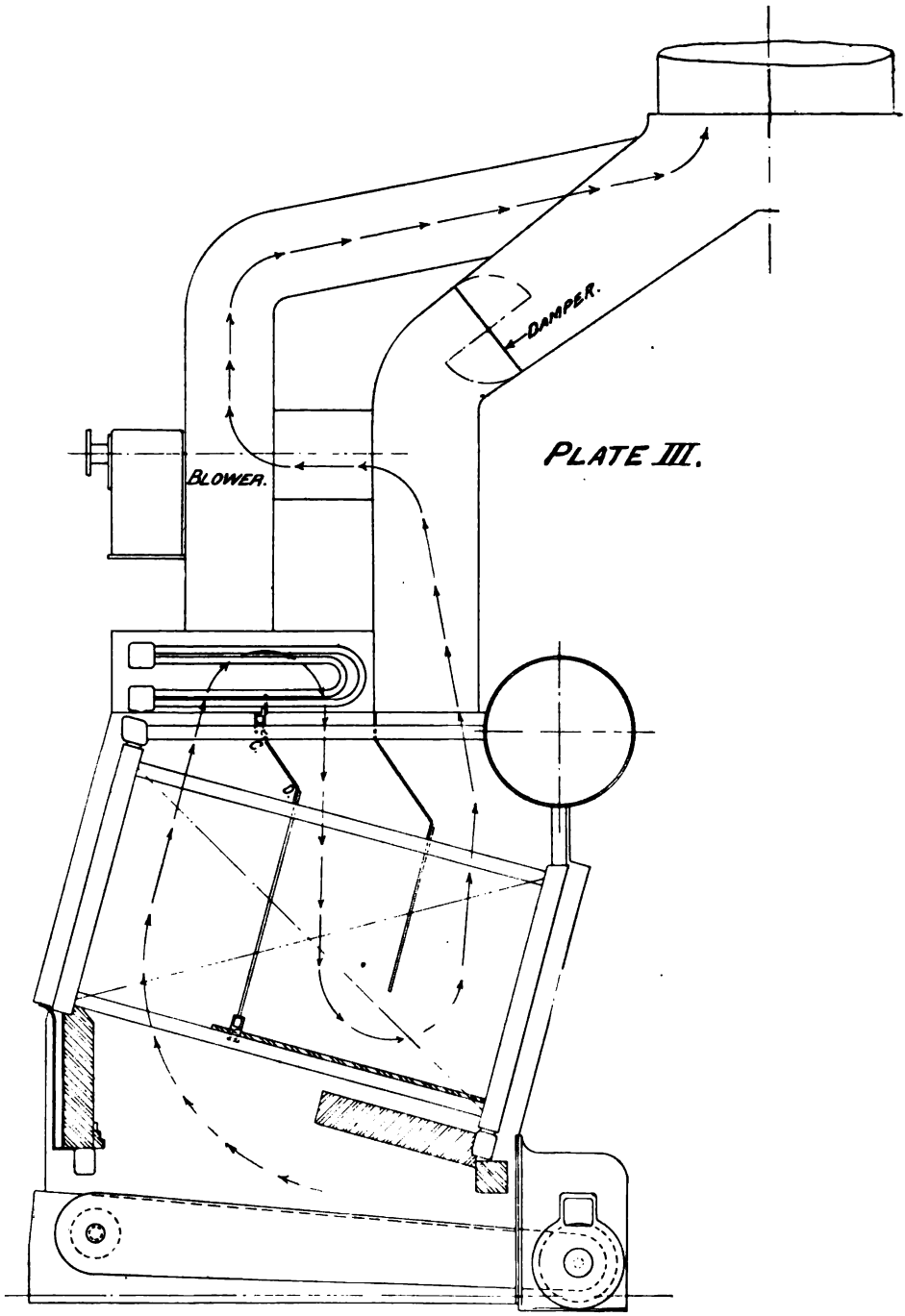

PLATE III.

INDICATING PATH OF GASES OF COMBUSTION WITH SUPERHEATER IN OPERATION AND USING INDUCED DRAFT.

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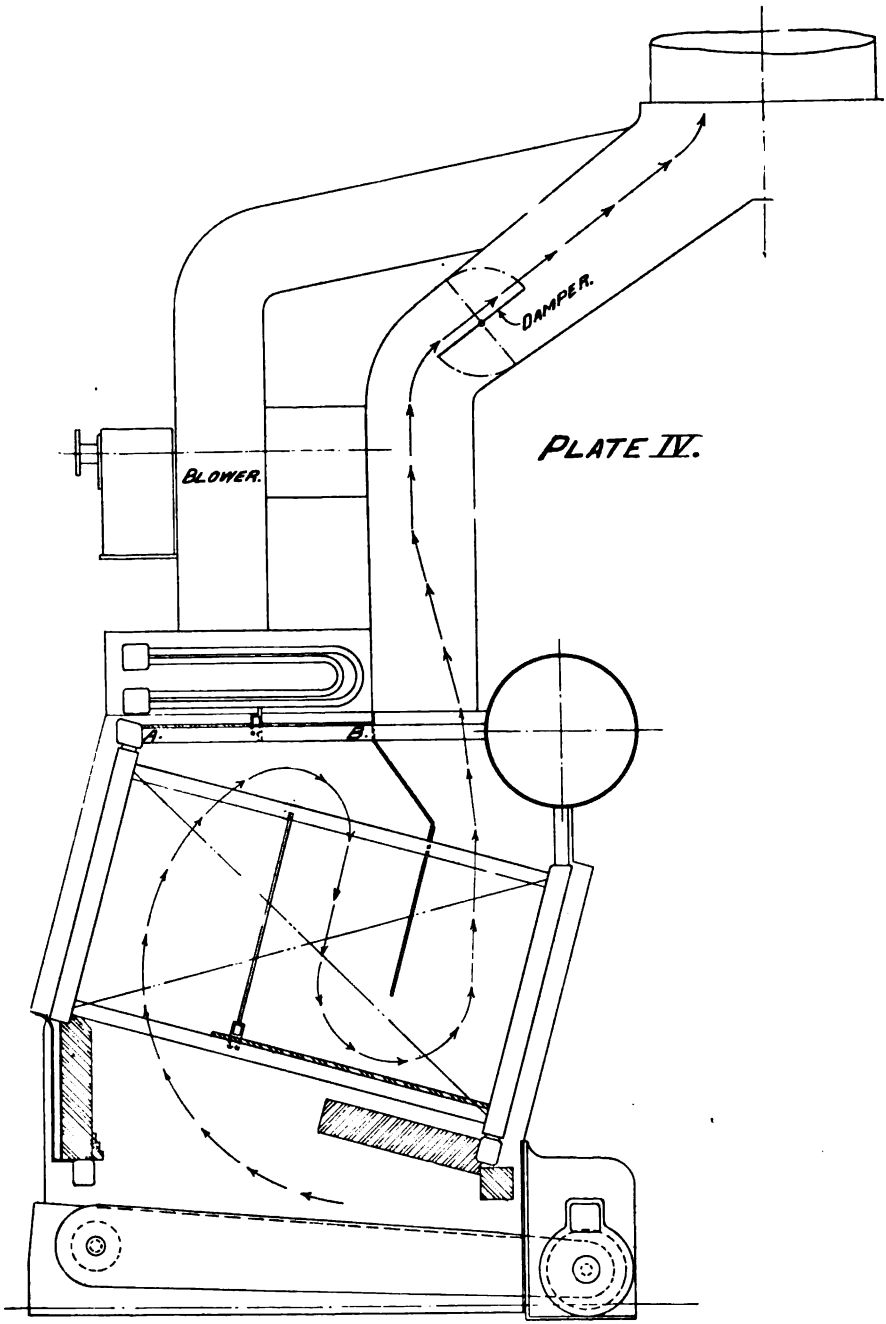

PLATE IV.

INDICATING PATH OF GASES OF COMBUSTION WITH SUPERHEATER CUT OUT AND BLOWERS STOPPED.

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Table I.—TEST OF MACHINERY OF STEAMER *JAMES C. WALLACE* (CONTROLLED BY ACME STEAMSHIP COMPANY), ON LAKE HURON, JULY 29, 1905, 9·30 A. M. TO 5·30 P. M., **USING SUPERHEATED STEAM.**

Duration of test, 8 hours. Draught, forward, 6 feet 4 inches; aft, 14 feet 8 inches. Speed, 12.6 miles.

Time.	Steam pressures per gauge.					Vacuum (inches).	Reading of engine counter.	Revolutions or double strokes per minute.						Temperatures, degrees, F.						M. E. Pressures.				I.H.P.					Coal.
	At boiler.	At engine.	1st receiver.	2d receiver.	3d receiver.			Main engine.	Main air pump.	Main feed pump.	Water-service pump.	Bilge pump.	Stroke, engine.	Steam at engine.	Steam leaving superheater. Port boiler.	Steam leaving superheater. St'b'd boiler.	Of gases at base of stack.	Injection.	Hotwell.	Feed.	H.P.	1st I.P.	2d I.P.	L.P.	H.P.	1st I.P.	2d I.P.	L.P.	Total.	
A. M.																													
9 30	237	233	93	34	10	24.5	154,819	...	25	18	48	16	134	478	484	483	...	66	104	194	...	...	...	...	...	...	...	...	...	
10·00	237	235	93	34	9.7	24.7	157,172	78.4	26	18	45	19	136	480	484	483	...	66	104	194	89	36	16.1	6.8	381	375.1	394.8	385.6	1,536.5	
10·30	237	234	94	34.5	10	24 6	159,531	78.6	26	16	48	18	140	474	484	483	487	66	104	192	91	36.2	16.2	6.9	390.6	378.1	398.2	392.3	1,559.2	
11·00	237	231	95	34	9.8	24.7	161,898	78.9	28	16	46	18	144	482	484	483	490	66	104	200	88.5	37	15.9	6.8	381.3	388	392.4	388	1,549.7	
11·30	237	233	93	34	9.7	24.6	164,254	78.5	28	15	48	19	142	484	486	486	485	66	104	198	91	36.8	15.6	6.7	390.2	383 9	383	380.4	1,537.5	
12·00	237	234	93	34	9.8	24.6	166,612	78.6	28	18	48	18	142	480	486	486	483	66	106	196	90	36.8	16.1	6.8	386.3	384.4	395 8	386.6	1,553.1	
P. M.																													
12·30	229	228	94	34	10	24.6	168,959	78 2	27	18	48	18	138	478	490	488	486	64	104	194	88.5	37	16	6.6	378	384 5	391.3	373.3	1,527.1	
1·00	237	230	94	34	10	24.4	171,321	78.7	27	18	48	18	140	479	489	488	480	64	104	209	88	36.5	15.9	6.5	378.2	381.7	391.4	370	1,521.3	
1·30	237	233	94	34	10	24.6	173,690	79	28	15	48	17	144	484	489	488	465	64	102	194	91.5	37	16	6.9	394.8	388.5	395.3	394.3	1,572.9	
2·00	237	231	94	34	10	24.5	176,066	79.2	27	17	48	17	144	484	489	488	481	64	104	194	89	36	16	6.8	385	378.9	396.4	389.5	1 549.8	*
2·30	237	234	94	33	10	24.5	178,431	78.8	25	16	46	17	142	483	489	488	493	63	104	195	89	35	16.1	6.75	383	366 5	396.8	384 7	1,531	
3·00	237	232	94	34	10	24.5	180,807	79 2	27	16	48	16	144	484	489	488	474	61	102	192	91	35.6	16.2	6.8	393.6	374 7	401.3	389.5	1,559.1	
3·30	237	234	95	34	10	24.5	183,170	78.8	27	16	48	16	144	487	487	488	...	62	102	193	91.5	36	16 2	6.8	393.8	377	399.3	387.5	1,557.6	
4·00	239	232	95	34	10	24.5	185,563	79 8	26	16	48	17	144	484	489	488	502	63	104	194	91	35 5	16	7	396.6	376.5	399 4	404	1,576.5	
4·30	242	240	96	35	10.2	24.5	187,956	79.8	26	16	48	16	140	486	490	489	...	63	105	196	93	36.5	16.1	7.3	405.3	387.1	401.8	421.4	1,615.6	
5·00	242	240	96	35	10.2	24.6	190,355	80	25	16	48	16	138	485	489	488	498.5	63	104	190	93.5	38	16.5	7.2	408.5	404	412.9	416 6	1,642	
5·30	242	239	95	34	10	24.6	192,749	79.8	26	16	46	16	138	486	489	488	...	62	104	192	...	...	...	...	...	...	...	...	...	
Total.	4,040	3,973	1,602	579.5	169.4	417.5	37,930	1,264 3	452	281	807	294	2,394	8,198	8,287	8,273	5,824.5	1,089	1,765	3,317	1,355 5	545.9	240.9	102.65	5,846.2	5,728 9	5,950.1	5,863.7	23.388.9	
Av'ge	237.6	233.7	94 2	34.1	9.96	24.5	79 02	79.02	26.6	16.5	47 5	17.17	140.8	482.2	487 4	486.6	485.3	64 06	103.8	195 1	90.36	36.39	16.06	6.84	389.7	381.9	396.7	390.9	1,559.2	

* Total coal (as weighed) = 18,900 pounds. Average coal (as weighed), per hour = 2,362.5 pounds. Moisture in coal, per cent. = 3.38. Ashes in coal, per cent. = 19 87.

Average number of degrees superheat at engine..... 81.7° F.
Average I.H.P. during test (main engine only)..... 1,559 2
Pounds of coal per I.H.P. per hour, as fired..... 1.515
Pounds of coal per I.H.P. per hour, dry..... 1.464

Average draft (inches of water) at base of smoke pipe, = 0.22
Average temperature engine room, lower platform, = 79° F.
Average temperature fireroom, = 81° F.
Test made under natural draft—fireroom blowers stopped.

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Table II.—TEST OF MACHINERY OF STEAMER *JAMES C. WALLACE* (CONTROLLED BY ACME STEAMSHIP COMPANY), ON LAKE SUPERIOR, JULY 30, 1905, 10 A. M. TO 4 P. M., **USING SUPERHEATED STEAM.**

Duration of test, 6 hours. Draught, forward, 6 feet 4 inches; aft, 14 feet 8 inches. Speed, 12.7 miles.

Time.	Steam pressures per gauge.					Vacuum (inches).	Reading of engine counter.	Revolutions or double strokes per minute.							Temperatures, degrees, F.							M. E. Pressures.				I.H.P.					Coal.
	At boiler.	At engine.	1st receiver.	2d receiver.	3d receiver.			Main engine.	Main air pump.	Main feed pump.	Water-service pump.	Bilge pump.	Stoker engine.	Fireroom blowers.	Steam at engine.	Steam leaving superheater. Port boiler.	Steam leaving superheater. St'b'd boiler.	Of gases at base of stack.	Injection.	Hotwell.	Feed.	H.P.	1st I.P.	2d I.P.	L.P.	H.P.	1st I.P.	2d I.P.	L.P.	Total.	
A. M.																															
10'00	244	239	97	35	10.2	24.3	253,200	...	24	16	24	22	132	168	488	493	490.5	510	60	114	205	89	35.9	16.2	7	...	...	...	...	...	.....
10'30	244	238	95	34	9.8	24.8	255,582	79.4	25	17	46	20	132	144	492	496	493.5	.....	58	105	195	90.1	35.6	16.2	6.9	390.7	375.6	402.3	396.3	1,564.9	.....
11'00	240	244	96	35	10	24 7	257,969	79.6	26	18	48	33	132	85	494	496	493.5	509	58	104	196	91.5	36.1	16 45	6 9	397.8	381.9	409.6	397.4	1,585.7	.....
11'30	244	239	95	34	9.8	24.8	260,356	79.6	25	17	40	32	130	136	491	496	495.5	.....	58	102	197	91.5	35.8	16.4	6.9	397.8	378.7	408 3	397.3	1,582.1	.....
12'00	242	239	95	34	9.8	24.9	262,744	79.6	24	15	34	30	118	120	491	498	495.5	487	54	99	196	91.5	35 2	16.4	6.95	397.8	372.4	408 4	400.2	1,578.8	.....
P. M.																															
12'30	235	239	95	34	9 5	24.8	265,124	79.3	27	17	35	30	128	136	491	496	493.5	492	55	102	198	93	35.6	16.7	6.9	402.7	375.2	415.8	395.8	1,589.5	.....
1'00	240	244	95	34	9.5	24.9	267,493	79	27	16	38	32	132	140	490	494	491.5	.....	53	98	200	92	35.9	17.2	6.95	396.9	376.9	425	397.2	1,596	.....
1'30	242	237	94	34	9.5	25.1	269,878	79.5	27	15	38	30	134	126	488	494	493.5	493	50	95	190	89	35.6	16 7	6 95	386.4	376.2	415.3	399.7	1,577.6	*
2'00	241	242	96	34	9.8	25.2	272,273	79.8	26	16	36	30	126	110	492	496	493.5	.....	46	92	193	91	35.6	16.5	7.1	396.6	377.6	411.8	409.8	1,595.8	.....
2'30	238	238	95	34	9.5	25.2	274,668	79.8	26	14	34	30	124	136	493	498	495.5	512	45	91	192	91	35.5	16.4	7.1	396.6	376.5	409.4	409.8	1,592.3	.....
3'00	242	245	96	35	10	24.8	277,061	79.8	25	15	35	30	125	150	492	500	496.5	.....	43	103	198	92	35.8	16.8	7.1	400 9	379.7	419.3	409.8	1,609.7	.....
3'30	242	243	96	35	9 8	25	279,460	80	26	13	36	30	126	136	489	498	495.5	490	44	103	200	91	36	16 6	7	397.6	382.8	415.4	405.1	1,600.9	.....
4'00	242	238	95	34	9 5	25	281,854	79 8	26	16	35	30	124	144	490	498	495.5	504	43	102	192	93	35.5	16.7	7	405.3	376.5	416.8	404.1	1,602.7	.....
Total,	3,136	3,125	1,240	446	126.7	323.5	28,654	955 2	334	205	479	379	1,663	1,731	6,381	6,453	6,423.5	3,997	667	1,310	2,552	1,185.6	464.1	218 25	90.75	4,766.1	4,530	4,957.4	4,822.5	19,076	
Av'ge	241.2	240.4	95.4	34.3	9 74	24.88	79 6	79 6	25.7	15.7	36 8	29.15	128	133.1	490.8	496.4	494.1	499 6	51.3	100.77	196 3	91.2	35.7	16.78	6 98	397.1	377.5	413.11	401.9	1,589.6	

* Total coal (as weighed) = 15,700 pounds. Average coal (as weighed) per hour = 2,616.7 pounds. Moisture in coal, per cent. = 2.89. Ashes in coal, per cent. = 15.32.

Average number of degrees superheat at engine..... 88° F.
Average I.H.P. during test (main engine only)..... 1,589.6
Pounds of coal per I.H.P. per hour, as fired..... 1.646
Pounds of coal per I.H.P. per hour, dry..... 1.598

Average draft (inches of water) at base of smoke pipe = 0 42
Average temperature engine room, lower platform, = 77° F.
Average temperature fireroom, = 76° F.
Test made with induced draft—fireroom blowers in operation.

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Table III.—TEST OF MACHINERY OF STEAMER *JAMES C. WALLACE* (CONTROLLED BY ACME STEAMSHIP COMPANY), ON LAKE SUPERIOR, AUGUST 3, 1906, 8 A. M. TO 4 P. M., **USING SATURATED STEAM.**

Duration of test, 8 hours. Draught, forward, 19 feet 4 inches; aft, 19 feet 5 inches. Speed, 11.9 miles.

Time.	Steam pressures per gauge.					Vacuum (inches).	Reading of engine counter.	Revolutions or double strokes per minute.						Temperatures, degrees, F.				M. E. pressures.				I.H.P.					Coal.
	At boiler.	At engine.	First receiver.	Second receiver.	Third receiver.			Main engine.	Main air pump.	Main feed pump.	Water-service pump.	Bilge pump.	Stoker engine.	Of gases at base of stack.	Injection.	Hotwell.	Feed.	H.P.	First I.P.	Second I.P.	L.P.	H.P.	First I.P.	Second I.P.	L.P.	Total.	
A. M.																											
8:00	237	236	107	41	11.8	24.5	492,093	...	20	24	27	12	150	...	50	107	185	82.5	39.5	19.5	8	...	...	...	...	...	
8:30	237	238	108	41	12	24.5	494,485	79.7	19	23	34	14	144	554	50	105	185	83	40.5	19	8.05	361.3	429	473.7	464.1	1,728.1	
9:00	237	237	107	41	12	24.8	496,884	80	18	17	40	13	145	...	46	97	175	82.5	39	19	8.1	360.4	414.7	475.4	468.7	1,719.2	
9:30	235	234	108	41	12	24.5	499,280	79.9	19	20	36	13	145	548	46	104	184	81	41.5	19.1	8	353.5	440.7	477.4	462.4	1,734	
10:00	235	229	106	40	11.5	24.6	501,671	79.7	21	12	38	12	120	...	43	104	178	80	40.5	19.2	8	348.2	429	478.6	461.2	1,717	
10:30	239	235	108	41	12.1	24.6	504,064	79.7	20	18	38	12	144	549	43	106	187	82.5	41.2	19.2	8.1	359.1	436.4	478.6	467	1,741.1	
11:00	237	233	106	41	12	24.6	506,460	79.9	20	12	36	12	144	...	43	105	191	80.1	40	19	8.2	349.6	424.8	474.9	473.9	1,723.2	
11:30	237	234	101	37	11	24.6	508,882	80.7	24	14	36	11	142	584	42	106	186	88	41	19	7.5	387.8	439.7	479.6	437.8	1,744.9	
12:00	235	230	103	38	11	24.5	511,288	80.2	20	16	35	11	142	...	42	105	180	88	40.6	18	8	385.5	432.7	451.5	464.2	1,733.9	
P. M.																											
12:30	239	237	97	36	11.7	24.6	513,711	80.8	19	18	38	11	140	557	40	104	182	90	41.5	18	8.2	397.2	445.6	454.9	479.2	1,776.9	
1:00	239	242	95	35	11.2	24.8	516,127	80.5	20	18	36	10	140	...	40	104	183	93	42.5	18	8.2	408.9	454.7	453.2	477.5	1,794.3	
1:30	241	233	95	35	11.5	24.5	518,554	80.9	21	20	38	11	144	578	41	105	180	86	40.5	17.5	7.9	380	435.5	442.8	462.3	1,720.6	
2:00	237	236	97	36	11.6	24.5	520,982	80.9	20	19	38	11	144	...	41	105	184	88.5	42	18	8.2	391	451.6	455.5	479.8	1,777.9	
2:30	232	226	91	33	10.7	24.6	523,409	80.9	20	20	42	10	142	534	42	104	179	88	40	17	7.9	388.8	430.1	430.2	462.3	1,711.4	
3:00	237	234	95	35	11.5	24.5	525,804	79.8	22	21	34	11	142	...	43	106	184	88	41.5	17.8	8.2	383.5	440.2	444.3	473.3	1,741.3	
3:30	242	240	98	37	12.1	24.4	528,224	80.7	21	21	36	12	136	553	44	109	190	88	41.5	18	8.4	387.9	445.1	454.4	490.4	1,777.8	
4:00	239	235	98	37	12.1	24.5	530,657	81.1	21	21	36	11	136	...	44	108	184	91	41.5	18.2	8.5	403	447.3	461.6	498.6	1,810.5	
Total.	4,035	3,989	1,720	645	197.8	417.6	38,564	1,285.4	345	314	618	197	2,400	4,457	740	1,784	3,117	1,460.1	694.8	313.5	137.45	6,045.7	6,997.1	7,386.6	7,522.7	27,952.1	*
Av'ge.	237.3	234.6	101.17	38	11.63	24.56	80.34	80.34	20.3	18.47	36.94	11.59	141.17	557.1	43.53	105	183.3	85.9	40.87	18.44	8.08	377.8	437.3	461.6	463.8	1,740.5	

* Total coal (as weighed) = 25,100 pounds. Average coal (as weighed) per hour = 3,137.5 pounds. Moisture in coal, per cent. = 3.07. Ashes in coal, per cent. = 17.8.

Average I.H.P. during test (main engine only).....1,740.5
Pounds of coal per I.H.P. per hour, as fired..... 1.803
Pounds of coal per I.H.P. per hour, dry..... 1.747
Ship loaded with 10,400 tons (2,240 pounds) of iron ore.

Average draft (inches of water) at base of smoke pipe = 0.24
Average temperature engine room, lower platform, = 70° F.
Average temperature fireroom, = 73° F.
Test made under natural draft—fireroom blowers stopped.

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Table IV.—TEST OF MACHINERY OF STEAMER *JAMES C. WALLACE* (CONTROLLED BY ACME STEAMSHIP COMPANY), ON LAKE HURON, AUGUST 4, 1906, 8·15 A. M. TO 2·15 P. M., **USING SATURATED STEAM.**

Duration of test, 6 hours. Draught, forward, 19 feet 4 inches; aft, 19 feet 5 inches. Speed, 11.2 miles.

Time.	Steam pressures per gauge.					Vacuum (inches).	Reading of engine counter.	Revolutions or double strokes per minute.						Temperatures, degrees, F.				M. E. pressures.				I.H.P.					Coal.
	At boiler.	At engine.	1st receiver.	2d receiver.	3d receiver.			Main engine.	Main air pump.	Main feed pump.	Water-service pump.	Bilge pump.	Stoker engine.	Of gases at base of stack.	Injection.	Hotwell.	Feed.	H.P.	1st I.P.	Second I.P.	L.P.	H.P.	1st I.P.	2d I.P.	L.P.	Total.	
A. M.																											
8·15	239	240	99	37	10.5	24.4	585,601	...	18	15	54	34	146	...	63	106	194	81	37.5	19	7 25	...	...	...	...	...	
8·45	239	242	97	36	10.3	24.5	587,961	78.7	23	16	52	37	144	572	63	104	193	84	35.7	17.4	7.25	361	373.4	428.4	412.7	1,575.5	
9·15	238	240	89	34	10.2	24.6	590,307	78.2	18	15	52	35	124	...	63	104	192	81	35.4	16.9	7.2	345.9	367.8	413.4	407.3	1,534.4	
9·45	239	243	89	33	10.1	24.7	592,651	78.1	19	18	50	36	126	554	63	103	192	80	35.6	17 2	7.2	341.2	369.5	420.1	406 7	1,537.5	
10·15	240	239	89	33	10	24.7	594,996	78.2	20	20	51	36	130	...	63	104	191	80	35.8	17.1	7.1	341.6	372	418.3	401.6	1,533.5	
10·45	239	243	89	33	10	24.7	597,321	77.5	20	16	44	40	126	558	63	104	191	83.5	36	17.1	7.15	353 4	370.8	421.8	400.8	1,546.8	
11·15	239	238	89	33	10	24.4	599,596	75.8	20	15	36	42	104	...	63	104	190	81.5	35 9	17.1	7.15	337.4	361.6	405.5	392	1,496.5	
11·45	239	237	89	33	10.1	24.5	601,874	75.9	24	16	36	38	126	532	64	104	195	84	38.1	17	7 1	348.2	384.3	403.6	389.8	1,525.9	*
P. M.																											
12·15	239	236	89	33	10.1	24.6	604,192	77.3	26	18	36	38	134	...	64	104	195	83.5	36.5	17	7 1	352.5	375	411	397	1,535.5	
12·45	239	236	89	33	10.1	24 7	606,504	77.1	25	15	42	36	126	544	64	104	194	84	35.6	17	7.1	353.7	364.8	410	396	1,524.5	
1·15	237	235	89	33	10.1	24 6	608,785	76	22	16	40	38	126	...	64	104	192	85	38.5	17	7.2	352.8	388.9	404.1	395 8	1,541.6	
1 45	237	230	87	32	9.8	24.7	611,059	75.8	22	16	36	38	120	561	64	104	195	84	35 7	17	7	347.7	359.6	403 1	383.8	1,494.2	
2·15	239	233	89	33	10.2	24.6	613,324	75.5	22	15	38	38	120	...	63	104	192	84.5	36.5	17.4	7.25	348.4	366 2	410.9	395.9	1,521.4	
Total,	3,103	3,092	1,173	436	121.5	319.7	27,723	924.1	279	211	567	486	1,652	3,321	824	1,353	2,506	1 ,076	472.8	224.5	93 05	4,183.8	4,453.9	4,950.2	4,779.4	18,367.3	
Av'ge	238.7	237.8	90.2	33.5	10.12	24.6	77.01	77.01	21.46	16.23	43.61	37.4	127.1	553.5	63.4	104.1	192.7	82.77	36.37	17.27	7.16	348.6	371.1	412.5	398.3	1,530.5	

* Total coal (as weighed), = 16,200 pounds. Average coal (as weighed), per hour = 2,700 pounds. Moisture in coal, per cent. = 2.88. Ashes in coal, per cent. = 14.4.

Average I.H.P. during test (main engine only).....1,530.5
Pounds of coal per I.H.P. per hour, as fired..... 1.764
Pounds of coal per I.H.P. per hour, dry..... 1.713
Ship loaded with 10,400 tons (2,240 pounds) of iron ore.

Average draft (inches of water) at base of smoke pipe, = 0.22
Average temperature engine rooom, lower platform, = 84° F.
Average temperature fireroom, = 79° F.
Test made under natural draft—fireroom blowers stopped.

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SUMMARY OF TESTS.

	Test I.	Test II.	Test III.	Test IV.
Duration, hours	8	6	8	6
Kind of steam	Super-heated.	Super-heated.	Satu-rated.	Satu-rated.
Steam pressures, gauge:				
At boilers	237.6	241.2	237.3	238.7
At engines	233.7	240.4	234.6	237.8
At first receiver	94.2	95.4	191.2	90.2
At second receiver	34.1	34.3	38	33.5
At third receiver	9.96	9.74	11.63	10.12
Vacuum, inches	24.5	24.9	24.6	24.6
Revolutions per minute	79	79.6	80.3	77
Cut off in cyls., per cent. of stroke:*				
H.P. cylinder	.574	.574	.574	.504
First I.P. cylinder	.659	.659	†.647	.698
Second I.P. cylinder	.598	.598	†.606	.64
L.P. cylinder	.605	.605	†.556	.58
Mean effective pressures:				
H.P. cylinder	90.36	91.2	85.9	82.77
First I.P. cylinder	36.39	35.7	40.87	36.37
Second I.P. cylinder	16.06	16.78	18.44	17.27
L.P. cylinder	6.84	6.98	8.08	7.16
Indicated horsepower:				
H.P. cylinder	389.7	397.1	377.8	348.6
First I.P. cylinder	381.9	377.5	437.3	371.1
Second I.P. cylinder	396.7	413.1	461.6	412.5
L.P. cylinder	390.9	401.9	463.8	398.3
Total	1,559.2	1,589.6	1,740.5	1,530.5
Coal:				
Moisture in per cent	3.38	2.89	3.07	2.88
Per hr., as fired, pounds	2,362.5	2,616.7	3,137.5	2,700
Per hr. per I.H.P. (moist), lbs	1.515	1.646	1.803	1.764
Per hr. per I.H.P. (dry), pounds	1.464	1.598	1.747	1.713
Per hr. per sq. ft. grate (moist), lbs	15.96	17.68	21.2	18.24
Per cent. of ashes in coal	19.87	15.32	17.8	14.4
Kind of draft	Natural.	Induced.	Natural.	Natural.
Draft (ins. water), base of smokpipe	.22	.42	.24	.22
Temp. at base of smokepipe, deg. F.	485.3	499.6	557.1	553.5

* As determined from indicator cards.

† At 11:02 A. M. cut offs changed as follows: first I.P., .705; second I.P., 671; L.P., .575.

At Duluth, while loading with iron ore, the baffle marked C–D (plate III) was removed from both boilers, and an additional fire-brick baffle laid on the upper row of 4-inch tubes, as shown on A–B (plate IV). Under the changed conditions, it will be noted (plate IV) that the products of combustion no longer circulated around the superheater tubes. By opening the communicating valves between drums and main steam line, and closing the valves to the superheater, saturated steam

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is thus sent to the engine. Tests III and IV were made under these conditions on the down-trip of the vessel.

With the superheaters in use, the temperature of the steam was taken just after leaving each boiler, and again near the throttle of the engine. These temperatures will be found recorded in the tables. The results of Test I show an average superheat at the boilers of 86.9 degrees and at the engine of 81.7 degrees. These figures, for Test II, become respectively 92.1 and 88 degrees.

To determine the moisture, during the tests with saturated steam, a Barrus throttling calorimeter, attached to the main steam pipe near the H.P. cylinder, was used. The results indicated that the steam was practically dry.

CONCLUSIONS.

Comparing the results obtained with superheated steam (Tests I and II), it will be seen that the power developed by the main engine differs only slightly in the two tests. The cut-off in each cylinder was precisely the same in both tests, and the only variation in the auxiliaries run worthy of note was the addition, in Test II, of two fireroom blowers and two small cooling pumps. On the basis of dry coal, the saving in fuel amounts to about 8.4 per cent. in favor of Test I. This difference is scarcely attributable alone to the steam consumption of the additional auxiliaries in Test II, and the result very likely was largely influenced by furnace conditions. The inference, in any event, is plain, viz: that for moderate rates of coal consumption with the boiler installation on the *Wallace*, natural is more economical than induced draft.

In the first three tests the H.P. cut-off was set at the same point. This resulted in Test III (saturated steam), in the development of considerably more power as compared with I and II. In order to approximate to the power shown by the latter, it was necessary to shorten the cut-off of the H.P. cylinder, which was done in Test IV.

By reference to Tests I and IV, it will be seen (excluding cylinder cut-offs) that these were made under like conditions,

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as nearly as possible, with the exception that in the first, superheated, and in the second, saturated, steam was used. A comparison, based on dry coal, shows a net saving in fuel, with superheated steam, amounting to 14.5 per cent. In considering this result it should be remembered that it represents the combined increased efficiency of machinery plant, the most important factor in which is the improved economy of the engine when running under superheated-steam conditions.

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NOTES.

SUBMARINE BOATS AND THEIR SALVAGE.

Broadly speaking, there are two classes of accidents that can happen to a submarine boat. First, the admission of water into the interior; secondly, an explosion. Both these have their counterpart in surface vessels—namely, collision and boiler or ammunition accidents. But the confined space and small reserve of buoyancy of the submarine boat intensifies the danger of the result to the crew. And it is this, rather than the frequency of accidents, that has occasioned public comment.

In the case of admission of water, no help can be expected from watertight bulkheads; for, obviously, to keep the vessel afloat each compartment must not have a greater capacity than the equivalent of the reserve of buoyancy; but further, for such compartments to be effective, only one—namely, the center one—can be allowed the same volume as the reserve of buoyancy, since it is not only necessary to float the boat, but also to afford egress to the crew, and therefore the boat must be maintained approximately on an even keel. To do this no compartment except the midship one must exceed in capacity half the reserve buoyancy of the boat; this limitation is necessary so as to allow of a corresponding compartment, the opposite end of the boat being flooded to maintain the horizontal trim. Such subdivision would mean at least thirty compartments, and none except the one in the center of the boat could be more than 3½ feet between the bulkheads. Of course, such a subdivision is an impossibility. It may, however, be argued that it would be well to have one or two compartments so as to confine the water to definite portions of the boat, and to leave a certain amount of space for the crew to

*Abstract of paper by Captain R. H. Bacon, R. N., D. S. O., read before the Institution of Naval Architects.

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inhabit in case of accidents. This is a very reasonable contention, but before accepting it we must consider the probabilities of an accident happening, the chances of such an accident occurring in waters where salvage is possible, and, further, the chances of successful salvage to save life under ordinary conditions of sea and weather; all these must be balanced against the ever-present inconvenience in daily life of having the boat divided into these three or four compartments. Such subdivision is not merely an inconvenience as regards control and communication, but for another reason it reacts against the efficiency of the boat when actually running on service. One marked feature in practice is the way in which spaces below a certain size affect the normal mental condition of the men. To run a boat successfully, everyone inside her should be in an absolutely normal condition; the increased constriction of vision, and the necessary accuracy of handling, argue even greater coolness than in ordinary vessels. For a man's mental condition to be normal, free living space is required; too confined surroundings react on the crew, and by producing a feeling of restriction create a mental tension; hence, unless some very strong reasons exist for making living spaces below more confined than absolutely necessary, they should be as large as the boat will permit.

THE PROBABILITIES OF ACCIDENTS.

Taking first the case of water entering the boat. Two broad conditions exist when this is possible. One, where water enters a hatch; the other, when it enters through a leak.

The former is by far the more probable cause of such an accident. Of the four serious accidents whereby boats have foundered in the last few years, in fact, all modern accidents—namely, the Russian *Delfin*, the French *Farfadet*, the *A1* and the *A8*—all have been due to water entering the hatch.

The fact of the hatch being the main source of weakness is very suggestive, and most reassuring as regards the safety of the boats, since with the practical elimination of this source of danger the main cause of accidents up to the present will

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be obviated; in fact, if no other remedies were taken, the attention attracted to this point should alone be sufficient to prevent accidents of this nature happening in the future. Many suggestions have been put forward for operating such a hatch automatically; but no matter requires more careful consideration than suggestions for the automatic or semi-automatic closing of vital hatches, since, if any portion of a man, either body, hands or arms, is in the way, and is caught by the hatch, the aperture can never subsequently be made watertight, and the danger to the boat is greatly intensified. The simplest method of making the hatch of a submarine boat automatic is by hinging it on the foreside, so that the pressure of the water with the boat going ahead closes it; but had this been the case in the *A8* there is little doubt that the captain, who was leaning across the tower, would have been caught, and every subsequent chance of stopping the water entering would have ended. The best method seems to be to hinge the hatch so as to open athwartships, so that with headway on the boat no direct pressure to open or close it is exerted by the water; but should the tower submerge, the lateral inrush will undoubtedly shut it. But let us for a moment analyze the chances of the water ever reaching the hatch.

The following is the broad outline of the previously-mentioned accidents, so far as we know from the general accounts available:

In the case of the *Delfin* the hatch was quite low, too large a number of men were in the boat, the ballast tanks were flooded while the hatch was open (an original and deadly error), and the boat sank deep until the water lapped over into the boat. Such an occurrence should without doubt never have happened. No structural provision could ever forestall such an accident.

In the case of the *Farfadet* the boat was apparently diving with the hatch improperly shut, and in attempting to close it fully the hatch opened.

In the case of the *A8* the boat was being run with about one-third her normal buoyancy at considerable speed when she

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dived under, and her hatch was not closed by those near it. Now all these cases are of a nature that should not occur again; there is nothing inherent in submarine boats to render any of them liable to recur.

In the case of the *Delfin*, systematic drill and the most elementary care would absolutely preclude a repetition of such an accident.

With the *Farfadet*, a redesigned closing method would allow of tightening the hatch with safety.

With the *A8*, the maintenance of full buoyancy when steaming fast would have prevented the water ever approaching the hatch 11 feet above the water line. So far, therefore, we may legitimately state that these accidents are not such as in any way to destroy confidence in the general system of submarine work.

We have now to consider the question of the admission of water due to collision or grounding.

As regards the former, a collision of any magnitude must be a most serious matter; the reserve buoyancy in the boats is so small that even a small leak must be a very difficult matter to deal with. Watertight compartments have been shown to be out of the question, and therefore, at all hazards, collisions must be avoided. This being the case, it is the natural duty of a submarine boat to get out of the way of any craft that may do her damage. Such is only wisdom on her part, and with one craft determined to avoid, the chances of collision are greatly reduced. There is one point, however, greatly in favor of submarine boats—namely, the enormous strength of their structure compared with surface boats of similar tonnage; this is of the greatest importance in all minor collisions. Two cases are on record where the boats have been struck by the propeller of a ship and have been none the worse, and many cases of collision with other minor surface craft have left the submarine absolutely unaffected, whereas the surface craft have been seriously damaged.

In the particularly sad case of the *A1*, the conning tower was struck by a large ship and the hatch slightly sprung open;

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the structure otherwise was undamaged. The boat then traveled the whole length under the ship's bottom, during which time sufficient water had leaked in to prevent the boat subsequently rising. A watertight hatch now fitted at the base of the tower provides a means of guarding against any moderate leakage from such damage to the conning tower. Of course, should the hull of the boat other than the conning tower be damaged anywhere above the center line, then grave danger exists, and no considerations of design can save the boat against a considerable leak in such a position. This, however, is the one and only source of real danger to a submarine boat in the event of collision; and the only reasons why it is greater in her case than in that of surface craft are the impossibility of fitting watertight compartments and the short time available before her buoyancy is destroyed. This naturally leads us on to the consideration of what the reserve of buoyancy should be in a submarine boat. It is not permitted to discuss the design of boats, but it is infringing no secret to point out that the increase of reserve of buoyancy in the surface condition means increase of skin area for the same submerged displacement, and therefore for the same submerged horsepower. Storage of electrical power is extremely weighty, being for equal radius of action over thirty times that of the surface propulsion; for this reason only it is advisable, therefore, to keep the reserve buoyancy of the boats as low as is consistent with sea-going safety. No boat can be safer at sea than a submarine boat, since it is as tight as a bottle. No increase of actual safety is obtained by small increases in the reserve of buoyancy—say, from 15 to 30 or 40 per cent.—which entail other great disadvantages without *pro rata* gain. Provided, therefore, that the advantage gained in safety in case of collision by small increases in the reserve of buoyancy is very slight, and that the reserve is such as to ensure sea-going safety, no object exists in reducing the efficiency of the boats from the fighting point of view, by increasing the surface reserve of buoyancy unnecessarily, and so decreasing their speed when submerged.

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The other cause of danger, namely, grounding, is not so very great. There are two distinct conditions: the first, merely striking the bottom, which is amply provided for by practically the whole under surface of the boats being double-bottomed, as well as by the fact of the boats being buoyant, and so very considerably reducing the effect of a grazing blow. Danger of injury to the hull from this cause is really non-existent. The other possibility is of a boat sticking its nose into the mud. The sloping nature of the under side of the fore part of the boat does not render such an occurrence probable, unless the boat were proceeding at a phenomenally large inclination, and even then the leverage exerted in getting rid of ballast, combined with the action of the propeller going astern, should exercise great influence in freeing the boat.

The last possible cause of water entering is from the boat diving to too great a depth and the hull being crushed, or some of the hull fittings being unable to stand the pressure. This argues either the failure of the diving rudders, which are made as simple and as strong as possible, and should, therefore, be above suspicion, or that too much water ballast was being carried, and, therefore, that the boat would sink from want of buoyancy. This actually occurred in the *Porpoise* in America. Fortunately the boat only sank in about 120 feet, the boat being built for 100 feet pressure. She, however, had the defect of having her tanks only constructed to stand a pressure of 25 pounds on the square inch, or, roughly, about 50 feet of water; the water in them could not, therefore, be blown out, since it was suicidal to attempt to admit the air pressure necessary to expel the water. The large pressure of water on the hull caused the boat to leak considerably, the electric pumps could not work against the pressure, and it was merely a race between the hand pump and the in-leakage of water whether she was ever raised to the surface or not. Providentially, the hand pump gained the day, and the boat floated. In any boat, every tank that can possibly be put in connection with the sea should be tested to the full hull-resisting pressure. An accident such as this could not possibly

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occur with moderate care or proper design. The rate of blowing water from tanks is purposely made so great that a sinking boat should be caught and made buoyant long before a dangerous depth is approached. One turn of two handles, which are always kept in hand when the boat is diving, immediately relieves the boat of a large quantity of ballast.

Of all these possible causes of boats foundering from taking in water, we may fairly claim that the only one even probable is that when the boat is injured by collision in the hull above the center line.

EXPLOSIONS.

The question of accidents from explosions inside the boat next claims our attention. Three causes of explosions only are possible, namely:

1. From an explosive mixture of petrol and air through a leak of liquid petrol into the boat.

2. From a mixture of hydrogen given off from the batteries during charge, or in discharge, if, in the case of accident, the battery is flooded with water.

3. From the explosion of a compressed-air reservoir.

As regards petrol, the main important consideration is that to cause an explosion, first a leakage of liquid petrol is necessary; and, secondly, a spark is required to ignite the mixture. A leakage can invariably be detected by smell, should it occur; but in a properly-designed system leaks should be practically non-existent.

Even with vapor in the boat, no direct danger exists, provided the boat is properly ventilated, and no switch is moved, or anything is done to cause a spark. As a matter of fact, in practice, the smell of petrol inside a boat is almost unknown. In the case of the accident to the *A5*, where a petrol explosion occurred, the cause of the leakage was a badly-packed gland of the petrol pump, the gland being screwed down metal to metal; but, in spite of one man being overcome by the petrol fumes, the main motor was started, and the sparks determined the explosion. Had the very explicit and simple

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regulations provided been carried out, no accident would have occurred. The boats are particularly free from all chance of accident with the engines at work, since the volume of air used every two minutes is equivalent in volume to that of the whole air of the boat; with such enormous ventilation going on, no chance of vapor accumulating is possible. A prevalent idea is that the petrol tanks are themselves a source of danger. A moment's consideration will show that no danger from this cause can exist; since, whenever a closed vessel has petrol inside it, the enclosed air is saturated with vapor, and is therefore non-explosive. The tanks are constructed so that leakage is impossible, and many other precautions are taken so as to minimise every source of possible leakage in the petrol system. The English boats have covered over 30,000 miles under their engines, and, with the exception of one small flash in an early boat, no explosion except that in the *A5* has occurred. This of itself is fairly good evidence of the safety of petrol when moderate care is exercised. The second possible cause of an explosion is due to hydrogen given off by the batteries in charging; but, as this operation is only carried out when the boat is opened up for ventilation, no danger exists. The formation of this gas, however, has considerable bearing on the installation of the battery, and limits the extent to which it can be made thoroughly watertight, for no risks can be run of allowing an explosive mixture being formed inside an air-tight space. The explosion which occurred in the *A8* two hours after sinking was most probably due to the formation of this gas.

The third cause—namely, the failure of the air reservoirs—is merely mentioned as a possibility; no greater danger from this cause exists on board a boat than on board an ordinary ship, where such accidents are unknown. The usual periodic tests should fully suffice to ensure these reservoirs being in a thoroughly safe condition.

We may, I think, from the above short account assume that danger to the boats from explosions is really small, and not greater in comparison than the dangers which have attended

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the introduction of increased boiler and gun power in the Navy as a whole.

SALVAGE.

The ploblems connected with the salvage of these boats differ from those usually met with, chiefly because the question of rapidity is, or may be, of vital importance as regards the possibility of saving life. This factor is usually absent from other salvage operations, and no question then exists about setting deliberately to work and conducting the operations upon sound principles. The danger anticipated by the old adage, "More haste, less speed," truly applies to this as well as all engineering work.

* * * * *

Their maneuver grounds are chiefly in the deeper waters, and it is therefore evident that considerations of rapid salvage are totally out of the question should foundering occur in the majority of the waters in which they work.

The suggestion, therefore, for building special salvage craft for dealing with submarine boats when the boats mainly operate in waters where rapid salvage is impossible, is one that cannot for a moment be practically considered, even should the probable frequency of accidents warrant such action, which, as a matter of fact, it does not.

The first question as regards salvage is to provide for the succor of the imprisoned crew. Hundreds of suggestions have been received on this subject; all have received careful consideration, but up to date not one has been capable of practical application. The labor of attending to the numerous suggestions is very great, but it is well worth the expenditure of trouble if only some day one single good idea is forthcoming; and really it is impossible but to be grateful to all who attempt, however crudely, to assist the work of the country.

Let us consider the two most common and plausible of these suggestions.

Before adopting any special appliance or constructional complication, suggested solely from the point of view of saving life in accidents, two considerations should be fulfilled:

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first, the fighting efficiency of the boat must not be seriously affected by it, since the boats are designed primarily for fighting, and not for meeting with accidents; and, secondly, that such arrangements, when introduced, shall be efficient against all reasonable accidents that may happen.

It is simple to design a boat to be safe against any particular accident; the impossibility lies in designing one safe against every accident. The commonest device for the release of a crew from a boat is that of providing an air lock and diving chamber, and a hatch that can be opened from the bottom of the air chamber, the pressure of air preventing the water entering when the hatch is open; the crew can then, one by one, dive out and come to the surface. Supposing a sufficient space devoted to such an arrangement, there are two considerations which militate against its success. The hatch must necessarily be in the bottom of the boat; but if an accident happens, it is "even chances," even in the uncommon case where the boat is undamaged, whether the boat rests with its fore or after end on the bottom. It is therefore an even chance whether the men would, or would not, be able to use this means of egress. Again, it must be assumed that water is present in the boat, otherwise there is no reason that she should not come to the surface, and if this water is present, it is again an even chance if the compartment can be got at from the interior of the boat. But, further, even supposing this chamber could be used, we have still to consider the effect on the crew of the variation in pressure by suddenly rising to the surface from a considerable depth. It is common knowledge that under pressure the gases absorbed in the blood get compressed, and that sudden release of such pressure allows them to expand, producing large bubbles of gas in the blood, which cause breaks in its continuity; this is productive of the most serious effects on the system, so that, even if the men were able to find egress, in many cases their lives would not be saved. Since, therefore, this system cannot be safely used in deep water, not at all on a soft bottom, and unless duplicated and placed at each end of the vessel, it is useless with a

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small amount of water in the boat, and quite valueless with a large amount, it is not a fitting on which it is worth wasting much valuable weight and space. Another common suggestion is to provide a detachable chamber into which the crew may climb, which can be freed, and then will float to the surface. A few figures will dispose of the practicability of this idea. Allowing 1½ square feet of space for each man, eleven men will require 16½ square feet. If the chamber were made cylindrical, the thickness would be ¼ inch. It would have to be ballasted to keep it upright when floating, and, moreover, must float with a certain reserve of buoyancy to allow of the top being open even in smooth water for the egress of the men. This means that the cylinder would have to be at least 6 feet in diameter and 7 feet to 8 feet high. This would be an enormous excrescence outside the boat, and greatly increase her submerged resistance, even if further fitted with fair waters to relieve its circular shape, although increasing its size. Its dimensions would quite preclude the possibility of carrying it inside. In addition, the securing arrangements and watertight doors would add considerably to its weight, and then in the end a collision—which is, after all, the most probable cause of accident—would be liable to put it out of gear. Anyone who really considers carefully the difficulties of providing reasonable methods of exit, for a crew will soon be convinced of the futility of such special arrangements. If I am betrayed into saying that some convenient form of diving dress seems to provide the greatest chance of ultimate success, I add an appeal to non-experts who have neither seen the open sea nor have any knowledge of diving dresses to spare us written suggestions on this subject.

Suppose a boat at the bottom in less than 120 feet of water, the difficulties of rapid salvage are immense. Friendly critics who look on the Channel as a smooth sheet of water would have their ideas rudely dispelled if they had to undertake practical salvage operations. The number of days when sea and swell are absent are remarkably few, the hours available, owing to tides, are scanty, and the amount of physical work

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that divers are capable of doing in deep water is remarkably little. These ever-present handicapping factors, combined with the hundred and one little difficulties and delays that crop up, soon cause hours to run into days, and difficulties, not known to the outside world, cause those who have never had such work to do to wonder where the difficulty lies. But it is useless to dwell further on this matter; let us pass to the actual salvage of a boat on the bottom.

A boat on the bottom can be in one of two conditions—either with her hull intact or with it damaged. In the former case the salvage is fairly simple, and merely consists in connecting a compressed-air pipe to a socket in the hull, admitting compressed air, and blowing the water out. I must, in passing, take the opportunity of thanking the many people who have taken the trouble to write, either privately or to the press, suggesting the provision of such a fitting. But, needless to say, this has been provided from the very first inception of these boats.

If, however, the boat is damaged, she can either be lifted bodily, or the leak may be stopped and the water blown out, when she will rise of her own buoyancy, or do so with but little assistance. In the first case a considerable weight will have to be lifted, and, providing the weather conditions are favorable, no difficulty should exist; but the operation cannot be done without considerable labor. We will imagine for a moment that there is a chance of saving life in the boat, although a leak exists. In this case it will be necessary to lift the boat not merely to the surface, but sufficiently above the surface to allow of the hatch in the conning tower being opened without the danger, when the air pressure is relieved, of letting the water rise in the hull sufficiently to drown the survivors. For this purpose center-line lifting bolts on the hull are not of much use, since the purchases shackled to them will come "two blocks" before the hull is above water. Strops made by large hawsers are more convenient, since the blocks can be kept at the side as low as the middle of the boat. Again, the blocks that must be used weigh close on half a ton

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each; these are most difficult for a diver to handle. I do not know if I am addressing many who have been practical divers; if not, I must ask you to take my word as an old diver as to the enormous difficulty of bringing an unwieldy weight of half a ton near to, and of shackling it on to, a fixed point, such as an eye-bolt. Bringing a hawser to the block is far easier, or even shackling the hawser to the block on the surface, and then sweeping it into place under the boat, is a far easier and more practical method. Fixed bolts, under certain conditions, such as that of a boat being very deep in the mud, may be valuable, but for rapid salvage their utility is very problematic. Another point to be kept in mind is that the strength of the lifting gear to be used should be the sum of the weight of water in the boat and any full tanks, plus the displacement she has to be lifted to; moreover, the slinging has to be distributed so as not to strain the boat locally, cause her to leak, and allow the imprisoned air to escape from the inside.

The fact of lifting the boat above water necessitates the use of more than one lifting vessel, so that the boat may rise between them. This necessitates mooring these craft in exact position before any work can be undertaken, otherwise the tide will vitiate all subsequent operations. After everything has been done that can be done on the surface, the whole rapidity of the subsequent work depends on the tides and on the weather. To attempt to shackle on blocks below, with the lighters rising and falling 2 feet or 3 feet, is impossible, and to take the weight of the lift, except on a smooth day, is equally impossible, on account of the strain that would be brought on all the gear. The sudden immersion of the lighters and the inertia of the submarine are, in a sea, bound to strain the gear many times more than the mere weight to be lifted. For all these reasons rapid salvage is almost an impossibility, however hard everyone may work. Whatever appliances are available, wind, sea and tide will always be the determining factors as regards rapidity of work.

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The mere salvage of a boat, when rapidity is not essential, does not, even when in deep water, present any great difficulty. It is, in fact, easier in the case of a strong structure, like that of a submarine, than with an ordinary ship. Moreover, the peculiar shape of the hull makes it possible to sweep hawsers underneath the ends even in waters where it would be impossible to work divers.

It has been suggested from some sources that the Admiralty should keep their own salvage plant for the general use of the Navy. This is no concern of mine, and one for the Admiralty only to consider; but if I had any weight in their councils, I would most strongly advise their doing nothing of the sort. Anyone who has been practically associated with salvage work knows that success depends largely on practical experience and constant practice. The mere plant is quite a secondary consideration, and easily obtained. Moreover, in all large salvage operations special gear has to be designed and built. Such appliances cannot be kept stored. The valuable asset to salvage companies is the experience of their divers and the constant practice that the mercantile marines of the world afford them. In such commercial enterprises the Navy can never have a share; and our own service also does not, thank goodness, provide sufficient accidents to keep the men thoroughly practiced.

In conclusion, I hope that I have been able to put impartially before you the relative safety of submarine vessels compared to that of surface craft. That the danger of the work is apt to be exaggerated, I think you will all concede. That it requires constant care goes without saying; but with the exercise of such care, no apprehension need exist of a larger percentage of accidents than in other branches of the service. That with the increase in numbers of the boats accidents will occasionally occur is undoubted, for wherever large quantities of energy are stored in an easily available state danger must exist; but that this is present to an exaggerated extent in the case of submarine boats is not a fact. Nowhere is the extent of possible danger known better than among

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those who man the boats, and nowhere would the idea of excessive liability to accident be more scouted.

There is, however, another class of danger that may exist, though at present it does not do so, and that is that, in the desire to avoid all risks, the efficiency of the boats may be reduced both by constructional limitations and also by curtailing their practical maneuvering by subjecting them to work less stringent than that which simulates war conditions. Much better have no boats at all than allow such limitations to creep in. At present we can safely say that all the work the boats do is up to the full requirements of war training, and that neither of the two lamentable accidents to the boats have in any way detracted from the severity of the tactical work. The sympathy of the whole country with the sad fate of those splendid men, who, being volunteers, were the pick of the service, is apt to lead to exaggeration of the real danger of the work. It is rather outside the Navy than in it that apprehension as to the safe use of the boats arises. If this paper has succeeded in putting the real facts in a more dispassionate light, I feel that the half hour you have so kindly afforded me will not have been wasted, so far as the interests of the country are concerned.—"Engineering," London.

TURBINES AND RECIPROCATING ENGINES IN THE MIDLAND RAILWAY COMPANY'S BOATS.*

For further description of these vessels see JOURNAL A. S. N. E., Vol. XVI, pages 629, 631, 713, 1023, 1339 and 1340.

In January, 1903, the Midland Railway Company decided to build four new screw steamers for their Irish and Isle of Man services, in view of the approaching completion of their new harbor at Heysham, in Morecambe Bay, Lancashire. They entrusted the designing of these steamers to my partner, Professor Biles, and myself. The method of propulsion was naturally one that came up for discussion.

The only vessels fitted with the Parsons marine turbine

* Paper read at the summer meeting of the Institution of Naval Architects by William Gray.

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system of propulsion at that time were two small passenger steamers, trading in summer on the smooth waters of the Clyde estuary. As compared with the ordinary Clyde steamers, the results obtained were highly satisfactory, although their maneuvering qualities left something to be desired. The London, Chatham and South Eastern Railway Company had a turbine steamer under construction, but it was necessary to place the contracts for the Midland steamers before the former was tried.

After careful consideration of the data available the Midland Railway Company decided to fit two of the vessels of their new fleet with reciprocating engines and two with Parsons marine turbines.

Three of the vessels—the *Antrim*, *Donegal* and *Londonderry*—were intended for the Belfast passenger and cargo trade, and the *Manxman* for the Isle of Man summer passenger traffic, although she was also fitted with portable cabins for taking up the Belfast trade in winter.

The principal dimensions of the three former vessels are as follows: Length on the water line, 330 feet; molded breadth, 42 feet; molded depth, 25 feet 6 inches. The *Manxman* is similar in form and of the same length and depth, but the molded breadth is 43 feet.

The boiler installation in all the vessels consists of two double-ended and one single-ended boiler, the principal dimensions of which are given in parallel columns:

	Double ended.	Single ended.
Number	2	1
Length	22 feet.	11 feet 6 inches.
Diameter	15 feet 7 inches.	
Number of furnaces (total)	12	3
Heating surface in each ship	12,500 square feet.	
Grate area	400 square feet.	

The working pressure in the *Antrim*, *Donegal* and *Manxman* is 200 pounds per square inch, and in the *Londonderry* 150 pounds per square inch.

Forced draft on the closed-stokehold system is fitted in

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all the vessels. The fans for this purpose are driven by electric motors in the *Antrim* and *Donegal*, and in the *Londonderry* and *Manxman* they are driven by steam engines.

The engines of the *Antrim* and *Donegal* differ only in detail. They consist of two sets of the four-cylinder triple-expansion type, each driving a three-bladed propeller. The cylinders are 23 inches, 36 inches and two of 42 inches, with a 30-inch stroke.

The auxiliary machinery in all the steamers is of the most modern type, and, as the pumps are driven independently, a better comparison is afforded of the performances of the propelling machinery.

The arrangement of the turbines in the *Londonderry* and *Manxman* differs only in detail, but the turbines in the *Manxman* are larger, as they were designed for 25 per cent. more power than the *Londonderry*. The *Manxman* is also fitted with a Parsons vacuum augmentor. In each vessel there are three turbines; one high pressure and two low pressure. With the latter are incorporated the reversing turbines that work *in vacuo* when not in use. Each of the three turbines drives a separate shaft and a three-bladed propeller. The low-pressure turbines are on the outer shafts and the high-pressure turbine on the center shaft. All the turbines in both ships were made by Messrs. Parsons' Marine Steam Turbine Company.

The propellers of the *Antrim* and *Donegal* are three-bladed, the *Antrim's* being built and the *Donegal's* solid. The *Antrim's* is a little smaller in diameter and coarser in pitch.

The propellers of the *Manxman* are as follows: center, 6 feet 2 inches in diameter, 5 feet 7 inches pitch; side, 5 feet 7 inches in diameter, 5 feet pitch. The *Londonderry's* are all the same—5 feet in diameter, 4 feet 6 inches pitch.

The conditions of the contract were that each vessel was to maintain a speed of 20 knots for six continuous hours with the double-ended boilers only under steam and 300 tons dead weight on board.

The results of the official trials were as follows:

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	Antrim.	*Londonderry.*	*Manxman.*
Speed in knots	20.6	21.6	22.65

With all the boilers in use the results were:

	Antrim.	*Londonderry.*	*Manxman.*
Speed in knots	21.86	22.36	23.12

The amount of water consumed was measured during the progressive trials by counting the strokes of the feed pumps, and the comparative results are given in the following table:

Speed in knots.	Gallons of water consumed per hour.		
	Antrim and *Donegal.*	*Londonderry.*	*Manxman.*
14	4,500	4,500	4,500
17	6,700	6,100	5,800
20	9,700	8,900	8,300
22	...	13,600	12,500
23	...	...	17,300

These figures throw some light on the relative economy not only of the two systems of propulsion at various speeds, but on the different arrangements in the two turbine steamers themselves. They show that at from 14 knots to 20 knots the turbine is more economical.

The maximum difference occurs between 19 and 20 knots, which is the working speed of the vessels on service. In the case of the *Londonderry* the decrease in water consumption amounted to 8 per cent., and in the case of the *Manxman* to 14 per cent., as compared with the *Antrim* and *Donegal.*

The service to and from Heysham and Belfast was opened on September 1, 1904, and since that date there has been one vessel plying each way every night except Sunday. Since May 1, 1905, the *Manxman* has been employed exclusively on the Isle of Man route; and the results, being obtained under different conditions, are not included in the comparisons.

Through the courtesy of the officials of the Midland Railway Company the log books of the four steamers have been placed at my disposal, and it is, therefore, possible to make

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some comparisons that may be of interest to those connected with ships. As the question under consideration is reciprocating engines *versus* turbines, those occasions when the *Antrim* and *Donegal* were running together have been omitted. The *Londonderry* and the *Manxman* have never been running together on this service.

An accident to the high-pressure turbine of the *Londonderry* that happened early in the month of November interrupts the records of that steamer for three months. The cause of the accident has never been definitely ascertained, but everyone is agreed that no fault can be attributed to the turbine as a means of propulsion. The dummy rings on the casing and drum were allowed to come into contact, and one-half of them were destroyed, but the circumstances were rather peculiar, and not likely to occur in normal conditions of working.

Table A gives the following particulars for the various steamers, viz:

The number of single trips made from Belfast to Heysham, and *vice versa.*

The number of tons of coal consumed while under steam.

The total number of hours under steam.

The total number of hours at full speed.

The amount of coal consumed in port varies according to circumstances, and as it does not materially affect the matter under discussion, it has been excluded from these tables. It amounts approximately to about eight tons per steamer per single trip.

The column "Hours under steam" gives the time from quay to quay, and includes the time occupied in maneuvering the vessel in harbor.

The table treats the matter from a purely commercial standpoint; that is to say, it ignores the differences of speed and coal consumption due to the relative weights of turbines and reciprocating engines, and only deals with the items that exclusively interest the shipowner. It gives the amount of coal that each vessel consumed on a given number of trips. In all four steamers the three boilers have been in use, and all have

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Table A.

Dates.	Number of passages.	T.-S.S. *Antrim.* (Reciprocating engines.)			T.-S.S. *Donegal.* (Reciprocating engines.)			*Londonderry.* (Turbines.)			*Manxman.* (Turbines.)		
		Tons of coal consumed.	Hours under steam.	Hours full speed.	Tons of coal consumed.	Hours under steam.	Hours full speed.	Tons of coal consumed.	Hours under steam.	Hours full speed.	Tons of coal consumed.	Hours under steam.	Hours full speed.
1904.													
Sept. 1 to 10	9	282	67.3	55.1	...	...	...	284.8	65.3	54.9	...	...	...
23 to 28	5	159.5	36.3	28.1	...	...	...	161	34.3	30.5	...	...	...
29 to Oct. 17	16	...	...	...	498.4	116.5	103.3	527.4	111	97.5	...	...	...
Oct. 18 to Nov. 1	13	447.5	89	71	...	...	...	429.3	92.2	75	...	...	...
Dec. 1 to 15	13	...	...	...	465	91.3	78.5	...	...	...	507.7	84.1	74.1
16 to 21	5	204.7	34	32.1	...	...	...	...	...	...	187	39.1	23.5
1905.													
Dec. 29 to Jan. 10	11	416.4	73.1	67	...	...	...	...	...	...	420.5	73.3	62.5
Jan. 25 to Feb. 8	13	...	...	...	520	89.8	76	...	...	...	532	92.1	59.6
Feb. 9 to 23	13	499.7	89	72.1	...	...	...	...	...	...	512	83.8	73.5
Mar. 11 to 25	13	...	...	...	522	86.8	77.2	...	...	...	530	86.2	70.7
31 to April 10	9	345.6	58	52.5	...	...	...	350.5	62.5	52.2	...	...	...
April 26 to May 10	13	...	...	...	551	89.2	78.4	512	84	72.4	...	...	...
May 12 to 25	12	472.8	74.5	68.2	...	...	...	470.3	77.4	67.5	...	...	...
June 10 to 24	13	...	...	...	459.1	87.1	78.5	510.5	82.9	72.9	...	...	...
	158	2,828.2	521.2	446.1	3,015.5	560.7	491.9	3,245.8	609.6	522.9	2,689.2	458.6	363.9

SUMMARY.

	Antrim.	*Donegal.*	*Londonderry.*	*Manxman.*
Number of passages	77	81	90	68
Coal per passage, tons	36.7	37.2	36.1	39.6
Hours at full speed per passage	5.78	6.07	5.81	5.35
Hours at full speed, percentage of total hours under steam	85.5	87.7	85.7	79.5

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been working with the same moderate air pressure in the stokeholds. The results, therefore, represent the amount of coal consumed under easy steaming in all circumstances.

To show the results obtained from each pair of steamers running simultaneously, but in opposite directions, the previous table has been split up, as shown in Table B.

Table B.—Showing Results Obtained by Steamers Running Simultaneously, but in Opposite Directions.

	Reciprocating engine.	Turbine.
	Antrim.	*Londonderry.*
Number of trips	48	48
Average coal per trip, tons	35.6	35.3
speed in knots	19.7	19.5
	Donegal.	*Londonderry.*
Number of trips	42	42
Average coal per trip, tons	36	36.9
speed in knots	19.2	19.8
	Antrim.	*Manxman.*
Number of trips	29	29
Average coal per trip, tons	38.6	38.6
speed in knots	19.5	20 3
	Donegal.	*Manxman.*
Number of trips	39	39
Average coal per trip, tons	38.7	40.2
speed in knots	19.3	20.3

The speeds obtained on the trial trips have been already dealt with, but, valuable as these results are, the conditions in which trial trips are conducted are more or less artificial, and shipowners are justified in not taking them as final guides on the subject.

The logs have been very carefully examined, and, neglecting those runs where full speed was not maintained for the whole time that the vessels were in the open sea, the results are as given in Table B.

These results point to a marked decrease in the coal consumption of the *Manxman*, as compared with the *Antrim* and *Donegal*. The *Manxman* did 20.3 knots for the same coal consumption that the *Antrim* had at 19.5 knots. A similar comparison of the *Manxman* with the *Donegal* gives nearly the same result.

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In other words, for a speed of 19.5 knots the *Antrim* requires 38.6 tons of coal, and the *Manxman* 35 tons—a saving of 9.3 per cent. The *Donegal*, for a speed of 19.3 knots, requires 38.7 tons, and the *Manxman* 35.4 tons—a saving of 8.5 per cent.

The performances of the *Londonderry* are nearly as efficient as those of the *Antrim*, but they are better than those of the *Donegal*.

They also indicate that the *Manxman*, with higher steam pressure, a smaller number of revolutions, and larger propellers, has done better than the *Londonderry*.

A further economy in the turbine steamers is effected in the amount of oil used for lubrication. The logs show that this amounts in both steamers to five gallons per single trip. This, again, permits of a further economy in the reduction of the engine-room staff from four greasers to two.

Speaking generally, therefore, the performances of the turbine steamers, especially the *Manxman*, have been greatly superior to those of the steamers fitted with reciprocating engines.

It is not possible to make a quantitative analysis of the cost of upkeep, but so far the turbines have cost practically nothing (excepting the cost of repairs due to the accident to the *Londonderry*), and they require very little attention compared with what is necessary in the very best running engines of the reciprocating type.

There can be no doubt that one great benefit derived from the use of turbines is the elimination of the vibration inseparable from the use of reciprocating engines in Channel steamers. The engines in the *Antrim* and *Donegal* were most carefully balanced, and the vibration at the service speed is almost imperceptible. The almost unanimous testimony, however, of those who have traveled in the vessels is that the turbine steamers are steadier.

The only real inferiority in the *Londonderry* and *Manxman* is the difficulty of maneuvering from rest in narrow waters. In this respect they compare unfavorably with the *Antrim* and

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Donegal. No doubt with further experience those in charge will become more expert in working with the small screws, but for turning from rest there is smaller power in the side propellers. Experiments were made at the trial trips that showed that the turbine steamers going full speed could be brought to rest in about a minute and a half. This is a good result, but actual experience has shown the relative inadequacy of the backing power starting from rest. There is no good reason why sufficient backing power cannot be obtained with small screws if the reversing turbines are made powerful enough.

Having dealt with the question of speed and coal consumption, there remain three other points in connection with the design, viz: space occupied, weight and cost.

The turbines occupied so much more floor space than the reciprocating engines that the electric plant had to be put in the tunnel.

Space may be saved on the upper decks, but this is not an unmixed advantage, as it was effected by reducing the light and air space to the turbine room. By our tonnage laws as at present, the actual volume of the light and air space is included in the gross tonnage; but 1¾ times the actual volume may be deducted in arriving at the net tonnage. The effect of decreasing the light and air space to the turbine room involved an increase in the net tonnage of nearly 50 tons.

	Gross tonnage.	Net tonnage.
Antrim	2,100	603
Londonderry	2,086	651

In the case of vessels trading between ports where tonnage dues are heavy this is a matter of great importance, and may balance the commercial gains in oil and fuel. In such cases the gain in space cannot be used with advantage.

The saving of weight in the turbine steamers is considerable. In the hull the weights involved, viz: engine seat, tunnel stools, boss frames and plating, &c., are reduced by 30 tons. Against this the buoyancy of the bosses is 29 tons less in the turbine than in the other steamers, so that from the draught

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point of view the saving in hull is neutralized by the loss of displacement.

The weight of engines, shafting and propellers in the *Antrim* and *Donegal* was 280 tons. The weight of the corresponding items in the *Manxman* was 195 tons, so that the total weight to be propelled is about 115 tons less, a difference of nearly 6 per cent. on the light weight of the steamer.

The difference of the initial cost of the turbines as compared with reciprocating engines is not great. Comparing the *Antrim* and the *Londonderry*, it amounted to 1½ per cent. of the total cost of hull and machinery.—"Engineering," London.

GAS ENGINE WITH PRODUCER FOR MARINE USE.

The following is an extract from an article in "Engineering," London, descriptive of the installation manufactured by Messrs. John I. Thornycroft & Co., Limited, London:

The vessel is built of steel, and is 60 feet long, 10 feet wide, and 2 feet 6 inches draught. She is a single-screw boat, the propeller working in a tunnel. There is a roomy cabin forward and another aft of the machinery space. Behind this again is a large watertight well, the steel floor of which is above the water level. There are self-acting freeing scuppers with hinge valves, on the life-boat principle. The fore peak is divided off by a watertight bulkhead, and the machinery space is between watertight bulkheads. Lavatory accommodation and other conveniences are provided.

The interesting feature of the vessel is naturally the novel type of machinery, which consists of a four-cylinder Thornycroft gas engine, on the Capitaine system. The cylinders are single acting, each 8¼ inches in diameter and 11 inches stroke, and develop collectively 75 brake horsepower at 300 revolutions per minute. In regard to general construction, the framing consists of mild steel plates connected by angle bars, thus forming a box-like structure of great rigidity, the steel plates being carried right round to form the crank pit. Each cylinder trunk is cast separately, and is contained in the fram-

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ing. The cylinder heads or combustion chambers are also separate castings, and are supported between plates riveted to the transverse members of the frame. Each cylinder trunk is attached to the head by means of four bolts. In this way the whole force of the explosion is taken by means of the plates riveted to the transverse members, and transmitted directly to the crank-shaft bearings, the latter being also bolted to the frame plates. The object aimed at is to do away with heavy bolts that would otherwise be needed to connect the cylinder head to the trunk, such as are usual with the ordinary construction of engines of this description. The cylinder-head joint is not, with this arrangement, likely to leak—a frequent source of trouble with the ordinary arrangement. Another advantage is that the cylinder head, which carries all valves and the ignition apparatus, may be readily removed for cleaning and inspection; the operation of disconnecting and replacing can be carried out in as short a time as six minutes. The importance of this feature will be appreciated by those who have had experience with suction gas plants.

All valves are actuated mechanically by means of cams, the cams being actuated by toothed gearing from the crank shaft. The valves open directly into the combustion chamber, thus minimizing the surface which would cool the gases. The cam shaft is hollow, and carries the spindle for actuating the ignition gear. This spindle is under the control of the governor, so that the spark is advanced or retarded according to the speed of engine required, the governor spring being hand-controlled at will. The governor also acts on a valve for throttling the mixture. The ignition is effected by a low-tension magneto, the magneto being on the after part of the engine frame, and driven by bevel gear from the governor shaft. The bevel wheel for driving the magneto is also controlled by the governor, so that the time of maximum current synchronizes with the time of ignition. A distributor is fitted, so that should one of the ignition plugs become short-circuited the working of the others would not be interrupted.

The cylinder heads are cast with jackets for water circula-

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tion, the cylinder trunks being fitted with a thin steel casing, water circulating between the cylinder and the trunk. Each cylinder head and each trunk has its separate service, which is at atmospheric pressure. By this arrangement it is possible to see whether the circulation is being properly maintained. The cooling water is circulated by a centrifugal pump, driven from the gas drier by a belt. There is forced lubrication for the cylinders, the bearings being lubricated by drip feed. There is an oil plunger pump, driven from the crank shaft, and this delivers oil to a mechanical distributor having eight pressure-feed connections and nine sight-feed connections. The gudgeon pins of the connecting rods are hollow, and serve as oil retainers, the oil being supplied by means of a syringe. This system of lubrication is economical, and has been found to act with certainty.

For starting the main engine an auxiliary petrol motor is used. This small engine is mounted on a pivoted base plate, and is swung over to tighten the belt when required to be used. When the main has been started, the belt of the starting engine is thrown off. Half-compression cams are fitted to the main engine to ease the starting. They are brought into play by hand, and released by the governor when the engine has attained its normal speed. Reversing is effected by means of epicyclic gear and a cone clutch, placed in the line of shafting, and forward of the thrust block. One hand wheel frees and reverses the engine in either direction.

The suction producer consists generally of a steel shell having a fire-brick lining surrounding the hot zone. The upper part forms a hopper for holding a supply of fuel, which descends automatically, as required. In the boat in question the supply will last for ten hours without recharging. Anthracite or coke is used. The fire bars are of channel section, to enable them to hold ash and to better withstand the intense heat. They are supported on cams, and may be lowered towards either side to facilitate clinkering. A steam generator is placed in the upper part of the producer. This consists of a shell with Field tubes, and serves the double

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purpose of cooling the cases and generating steam, which is decomposed in the fire in order to supply the necessary gases for the explosive mixture. It should be explained, however, that there is an alternative means of generating steam by the exhaust gases from the engine, and this is fitted to the boat in question. The gas is passed from the producer into a cooling tower, there being a spray of water from the top which falls by gravity; and there is also a finely-divided water jet injected into the bottom by means of compressed air. This serves to free the gases from dust or other objectionable matter.

For the effective working of the engine it is necessary that the gases should not contain moisture; and to free them from water a centrifugal dryer is fitted. The dryer also removes any trace of tar that may have escaped through the cooler. The drier runs at a high rate of revolutions, and is driven by a belt from the engine crank shaft. It forms an essential part of the apparatus, and is a distinguishing feature of the system. Two small centrifugal pumps are connected to the drier spindle. One of these, to which reference has already been made, is for supplying cooling water to the engine and gas cooler, the other being for drawing off the dirty water from the cooler and discharging it overboard. For mixing the air and gas there is a double-seated valve situated in the inlet pipe near the engine, air and gas being automatically regulated to the proper mixture to suit the speed of the engine; but for different qualities of gas the air regulation is by hand.

Preliminary trials have already been made with this system, and it has been found to give satisfactory results.

COMPARATIVE LOSSES IN THE RUSSO-JAPANESE WAR.

* * * * *

According to the figures given by the "Times" correspondent up to the date he states, Russia has lost 346,588 tons of fighting ships, not including minor craft; while the Japanese Navy will emerge from the fight with a displacement of 25,435 tons greater than when the war began. This latter figure, however, must be largely increased, as the latest information from

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Japan shows that a number of the large Russian vessels which were sunk have been raised, and they will be added to the Japanese Navy. The story of the Russian naval squadrons in Far Eastern seas since the war commenced is shown in the following table:

RUSSIAN NAVY.

Nature of vessels.	Engaged in the war.		Sunk.		Captured.		Interned.		Remaining.	
	Number.	Displacement.	Number.	Displacement.	Number.	Displacement.	Number.	Displacement.	Number.	Displacement.
		tons.		*tons.*		*tons.*		*tons.*		*tons.*
Battleships	15	180,980	12	144,958	2	23,110	1	12,912	...	...
Armored cruisers	7	63,533	5	38,979	...	...	...	...	2	24,554
Sea-going coast defence iron-clads	3	13,212	1	4,126	2	9,086	...	...	...	...
Cruisers	13	65,416	6	26,341	...	...	5	29,115	2	9,960
Other kinds, excluding converted cruisers	21	76,793	14	25,222	1	11,700	2	11,449	4	28,422
Destroyers	24	10,290	19	5,666	2	590	11	3,334	2	700
Totals	83	410,224	57	245,292	7	44,486	19	56,810	10	63,636

This table is not quite complete, as it does not include torpedo boats, special-service steamers, as well as converted cruisers and a number of vessels whose fate is uncertain. The broad result is very striking: out of a total of 83 ships, with a displacement of 410,224 tons, sent by Russia into the belligerent arena, only ten, with a displacement of 63,636 tons, remain in her fighting line.

The corresponding table for the Japanese Navy is to be found on page 879.

This table also does not include torpedo boats, special-service steamers and converted cruisers. A comparison of the two tables brings out the fact that the Japanese force of 76 vessels, with a displacement of 274,184 tons, has sunk or captured 64 ships, with a displacement of 289,778 tons. On the other hand, while the Japanese have lost 12 ships, with a total displacement of 46,025 tons, they have captured or raised more than double this amount of tonnage. From the news-

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papers received from Japan by last mail we see that the Japanese naval authorities call attention to the fact that the *Peresviet*, which has just been raised at Port Arthur, was one of the strongest ships in Russia's Pacific Squadron. She left Kronstadt in 1901.

JAPANESE NAVY.

Nature of vessels.	Engaged in the war.		Sunk.		Captured.		Interned.		Remaining.	
	Number.	Displacement.	Number.	Displacement.	Number.	Displacement.	Number.	Displacement.	Number.	Displacement.
		tons.		*tons.*		*tons.*		*tons.*		*tons.*
Battleships	6	86,299	2	27,757	...	...	...	...	4	58,542
Armored cruisers	8	74,178	...	...	...	...	...	...	8	74,178
Sea-going coast defence cruisers	2	11,112	...	...	...	...	...	...	2	11,112
Cruisers	20	68,481	4	12,733	...	...	...	...	16	55,748
Other kinds	18	26,689	4	4,797	...	...	...	...	14	21,892
Destroyers	22	7,425	2	738	...	...	...	...	20	6,687
Totals	76	274,184	12	46,025	...	...	...	...	64	228,159

The *Peresviet* and the *Ossliabya* were laid down in 1898, under Russia's new program of naval expansion, and were supposed to embody the best results of modern naval progress. The approval that was accorded to them by expert opinion led to the laying down of the *Pobieda*, a sister ship, in 1899. These vessels did not carry a very heavy armament—four 10-inch and eleven 6-inch guns. The "Hochi Shimbun" says that the *Poltava* will be the next vessel to be floated, and that she will be followed by the *Pobieda*. The *Poltava* is a smaller type of vessel, and somewhat older. Her displacement is 10,960 tons, and her speed only 16 knots. She is lying on an even keel. The *Pobieda* has a slight list, but it is not expected that any serious difficulty will attend her raising. The *Retvisan* (battleship, 12,700 tons, completed in 1900) and the *Pallada* (protected cruiser, 6,630 tons) are said to be the most injured, but the expectation is that they too will be saved, and in that event the *Sevastopol* alone would remain at the bottom. This forecast, if trustworthy, means that there will be added to the Japanese Navy three first-class and one second-class battle-

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ships, one first-class armored cruiser, and one protected cruiser. Adding the results of the battle in the Sea of Japan, and the salvage operations at Chemulpo, we get this table:—

SHIPS TO BE ADDED TO JAPANESE NAVY.

	Tons.
Orel, battleship, 1902,	13,516
Retvisan, battleship, 1900,	12,700
Peresviet, battleship, 1898,	12,674
Pobieda, battleship, 1900,	12,674
Poltava, battleship, 1894,	10,950
Nicholai I, battleship, 1889,	9,700
Bayan, armored cruiser, 1900,	7,800
Varyag, protected cruiser, 1899,	6,500
Pallada, protected cruiser, 1899,	6,630
Seniavine, coast-defence ironclad, 1894,	4,126
Apraxin, coast-defence ironclad, 1896,	4,126

It is stated that much of the success achieved in raising these ships is due to a new pumping machine recently imported. Two of these are at work. Each pumps 4,000 tons per hour. The name of Captain Sakamoto is specially mentioned in connection with the work. He commanded the unfortunate *Yashima*, and his exceptional skill in salvage operations has been conspicuous in the raising of the *Bayan*. These facts and figures speak volumes for the efficiency of the Japanese Navy, while the report of the Russian admiral reveals a state of affairs, both as regards men and material, which go a long way towards explaining the complete defeat of the Russian squadrons. The "Kokumiu Shimbun" remarks truly:—"Never was a Navy burdened with a heavier and more onerous responsibilty than the Imperial Japanese Navy at the beginning of the war. The enemy's Vladivostock and Port Arthur squadrons combined were in themselves a fair match for all the naval strength we could place under Admiral Togo's command; but, in addition, the enemy had the Baltic fleet to draw upon, while we had nothing more to fall back upon; so that Admiral Togo's work was not only to demolish a strong foe present—

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a task hazardous enough in itself—but to exterminate his reinforcements to come. How well Admiral Togo executed his part the results very clearly show."

The war has also been the means of adding considerably to the Japanese mercantile marine. The appended list gives the

	Ship's name.	Nationality.	Tonnage.
1.	*Ekaterinoslav*	Russia	5,627
2.	*Mukden*	Russia	1,567
3.	*Russia*	Russia	2,312
4.	*Argun*	Russia	2,458
5.	*Manjuria*	Russia	6,193
6.	*Alexander*	Russia	261
7.	*Resnick*, sailing ship	Russia	87
8.	*Nicolai*	Russia	123
9.	*Mikhael*	Russia	3,461
10.	*Kotic*	Russia	400
11.	*Juliade*	Russia	?
12.	*Manchuria*	Russia	2,937
13.	*Bobrick*, sailing ship	Russia	125
14.	*Nadejula*	Russia	68
15.	*Thalia*	Russia	?
16.	*George*	France	179
17.	*Fuhping*	Germany	1,393
18.	*Veteran*	Germany	1,199
19.	*Nigretia*	England	2,368
20.	*King Arthur*	England	1,416
21.	*Rosalie*	England	4,370
22.	*Redington*	England	4,421
23.	*Wilhelmina*	Holland	4,295
24.	*Bawtry*	England	2,407
25.	*Oakley*	England	3,798
26.	*Burmah*	Austria	3,071
27.	*M. S. Dollar*	America	4,216
28.	*Wyefield*	England	3,235
29.	*Siam*	Austria	3,160
30.	*Palos*	Germany	2,398
31.	*Apollo*	England	3,829
32.	*Scotsman*	England	1,677
33.	*Silvania*	England	4,187
34.	*Powderham*	England	3,019
35.	*Severus*	Germany	3,307
36.	*Romulus*	Germany	2,630
37.	*Easby Abbey*	England	2,963
38.	*Vegga*	Sweden	2,562
39.	*Venus*	England	3,558
40.	*Aphrodite*	England	3,949
41.	*Tacoma*	America	2,812
42.	*Harbarton*	England	3,256
43.	*Industrie*	Germany	198
44.	*Henry Bolckow*	Norway	1,006
45.	*Quang Nam*	France	1,431
	Total	....over...	108,000

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merchant ships captured by the Japanese Navy, and no doubt it will interest many of our readers. When not described as a sailing ship, steamer is to be understood. It is to be noted that in no case was a non-combatant vessel sunk by the Japanese Navy.

The Russian record of vessels captured is very different from that of the Japanese. It is not necessary to reproduce it, as it only contains the names of fifteen ships, which have an average tonnage of 234 tons, only one being over 1,000 tons.

* * * * *

—"Engineering," London.

TEST OF CURTIS STEAM TURBINE.

The data given below was obtained from the test of a 2,000-kilowatt Curtis turbine (900 revolutions per minute), made by Mr. Fredrick Sargent, of the firm of Sargent & Lundy, engineers, and Mr. Louis a Ferguson, vice-president and general manager of the Commonwealth Electric Company.

The machine is of the General Electric Company's standard four-stage type, designed in 1903, with a few changes, made as the result of experiments conducted during the past year. Other changes, to be embodied in future machines, but which could not be made in the one under test without rebuilding, are expected to give an even higher efficiency, and there seems to be no doubt but that the economy of these machines will equal that of any of the best European types.

On the strength of this test the Commonwealth Electric Company placed an order with the General Electric Company for two turbo-generators, each with a nominal capacity of 8,000 kilowatts and a maximum short-period capacity of 12,000 kilowatts.

All instruments used were carefully tested and standardized, and the surface condenser used showed practically no leakage.

Several preliminary tests gave results closely approximating those of the official one.

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DATA FROM TEST OF 2,000-KILOWATT CURTIS STEAM TURBINE, SCHENECTADY, N. Y., MAY 8, 1905.

FULL-LOAD TEST.

Duration of test, hours	1.25
Steam pressure (gauge), pounds	166.3
Back pressure (absolute), inches of mercury	1.49
Superheat, degrees Fahrenheit	207
Load, in kilowatts	2,023.7
Steam consumption per kw. hour, pounds	15.02

HALF-LOAD TEST.

Duration of test, hour	0.916
Steam pressure (gauge), pounds	170.2
Back pressure (absolute), inches of mercury	1.40
Superheat, degrees Fahrenheit	120
Load, in kilowatts	1,066.7
Steam consumption per kw. hour, pounds	16.31

QUARTER-LOAD TEST.

Duration of test, hour	1
Steam pressure (gauge), pounds	155.5
Back pressure (absolute), inches of mercury	1.45
Superheat, degrees Fahrenheit	204
Load, in kilowatts	555
Steam consumption per kw. hour	18.09

ZERO LOAD.

Duration of test, hours	1.33
Steam pressure, pounds	154.5
Back pressure (absolute), inches of mercury	1.85
Superheat, degrees Fahrenheit	156
Steam consumption per hour, pounds	1,510.5

In addition to the data obtained from the test made by Messrs. Sargent and Ferguson the following water rates were obtained for the same machine, running at 900 revolutions per minute and 175 pounds absolute pressure, under different conditions of superheat and vacuum:

Saturated	27½ inches	Water rate = 18.6
100 degrees superheat	27½ inches	Water rate = 17.15
150 degrees superheat	27½ inches	Water rate = 16.6
200 degrees superheat	27½ inches	Water rate = 16.1
Saturated	28 inches	Water rate = 18.1
100 degrees superheat	28 inches	Water rate = 16.7
150 degrees superheat	28 inches	Water rate = 16.15
200 degrees superheat	28 inches	Water rate = 15.65
Saturated	28.5 inches	Water rate = 17.6
100 degrees superheat	28.5 inches	Water rate = 16.3
150 degrees superheat	28.5 inches	Water rate = 15.75
200 degrees superheat	28.5 inches	Water rate = 15.25

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At 150 pounds absolute takes 103.5 per cent of these values.
At 200 pounds absolute takes 97.5 per cent.
At 215 pounds absolute takes 96 per cent.
At one-half load the water rate is approximately one pound higher.

LAP JOINTS AND BOILER EXPLOSIONS.

This journal has so frequently expressed its opinion that the most dangerous features in engineering works are generally the minor details, that it is hardly necessary to elaborate on the subject generally at this time. A specific instance of this truth is afforded, however, by the final report on an investigation of the Brockton boiler explosion, made a little while ago by the Hartford Steam Boiler Inspection and Insurance Co. The preliminary report of the company on this very disastrous accident was printed in the Current News Supplement of this journal on April 1, and it was then stated that the explosion was probably caused by a lap-joint crack. The boiler was frequently and carefully examined, yet it blew up on the day after it was placed in service, after an idle period, while it was carrying only its regular pressure of 80 pounds. It was 6 feet in diameter and 17 feet long, and built of ⅜-inch fire-box steel, made by Carnegie, Phipps & Co. The thorough examination made by the company's experts has led them to confirm their earlier statement of the cause of the explosion, and as the lap joint is used in about seven-eighths of the boilers now in service, it is manifest that it is a very important detail.

It is well known that in the passage of a plate through the bending rolls, for the purpose of giving it the proper form for a boiler, a great deal of strain in the metal is inevitably caused. If this were not the case the sheet would not be curved after the process, for steel is sufficiently elastic to resume its original form after being subjected to large stresses. Another fact that can be observed at any set of bending rolls is the lack of uniform curvature in a solid sheet at those edges where it enters and leaves the rolls. If these edges have been punched, the plate is often bent considerably along one of the lines of rivet holes. It is these edges which are brought together and

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riveted, and it is often necessary to force them together by sledging or its equivalent. In the case of a lap joint, the extra strains coming on the metal do not cease with the completion of the manufacturing processes, for when the boiler is put in service the tension in the two plates has a tendency to twist the joint in a way perfectly apparent to the reader who will make a sketch of such a joint. For all these reasons it is manifest that the steel used in a boiler must be selected with full regard for its ability to withstand these special stresses, due to the use of the lap joint, and the necessity for marked ductility in the plates, particularly those of considerable thickness for shells of small diameter, needs no demonstration.

Even with the best of care and materials a dangerous defect is liable to occur in the lap joint, which has received the name of the lap-joint crack. It was such a defect that caused the loss of 58 lives, injuries to 117 more people and the destruction of property valued at about a quarter of a million dollars. With such facts before one it is almost criminal to look upon the design of boiler details other than with extremely critical eyes, or to fail to study this peculiar class of cracks.

The lap-joint crack is defined by the Hartford company as a fracture of one of the plates, which follows the general course of the joint under the extreme edges of the rivet heads in such a position as to be entirely covered by the projecting lap of the unaffected plate. They have been discovered many times in this country and abroad, and in Great Britain some of the leading authorities on boiler construction have drawn attention specifically to their dangerous nature. The main fact to observe is that the crack, no matter which plate it may be in, always starts from that face of the affected plate which is in contact with the overlapping plate, progressing into the metal more and more deeply until the boiler is weakened perhaps to the point of explosion. It cannot be seen from either the outside or inside of the shell until it has penetrated clear through the metal or has run into a rivet hole, so that its presence is shown by escaping steam or minute cracks. Before such indications can be observed the boiler may have reached a condition of extreme

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danger. The inspectors of the Hartford company have found examples of such cracks running for long distances along the joint without perforating the metal at any point, and Mr. Fletcher, of the Manchester Steam Users' Association, has made a somewhat similar assertion concerning an examination he once made.

This danger has been so well known that some boiler designers and makers are urging the passage of legislation forbidding the use of boilers having lap joints. They assert that no boiler having butt joints has ever exploded, but neglect to add that the number of these boilers is very small compared with the other type. It is a question whether such a legal prohibition is advisable. A well-made lap joint in a boiler constructed of suitable steel is not essentially dangerous, although the result is by no means so good as a boiler of equal quality with butt joints. Instead of prohibiting the use of the lap joint it would be wiser for the present to hedge its use about with so many precautions that it will become as safe as such construction can be made.—"Engineering Record."

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SHIPS.

UNITED STATES.

Kansas—*Launch of.*—The U. S. battleship *Kansas* was successfully launched from the yard of the New York Shipbuilding Company, Camden, N. J., at noon on August 12th.

The construction of the *Kansas*, together with her sister ships, the *Vermont* and *Minnesota*, was authorized by an act of Congress in 1903.

The general characteristics of these vessels are as follows: Trial displacement, 16,000 tons; mean draught for trial displacement, 24 feet 6 inches; length on load-water line, 450 feet; length over all, 456 feet 4 inches; breadth, extreme, 76 feet 10 inches; speed, 18 knots.

The main battery will consist of four 12-inch, eight 8-inch, and twelve 7-inch breech-loading rifles. In the secondary battery there will be twenty 3-inch rapid-fire guns of 50 calibers in length, twelve 3-pounder semi-automatic, six 1-pounder automatic, two 1-pounder semi-automatic guns, two 3-inch field pieces, two machine guns of .30 caliber, and six automatic guns of .30 caliber.

For armor protection there will be a complete water line belt 9 feet 3 inches wide and 9 inches thick for a distance of about 285 feet amidships, with a reduction to 4 inches at the bow and stern; the armor for the lower casemate is to be 7 inches thick for the sides and 6 inches athwartships, while for the upper casemate the thickness is 7 inches for both sides and athwartships.

For the 12-inch guns the turret armor is 12 inches at front, 8 inches on sides and rear, and 2½ inches on top; that for the barbettes is 10 inches at front and 7½ inches at rear above the gun deck, and 6 inches throughout between the gun and protective decks.

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In the case of the 8-inch guns, the turret armor is 6½ inches at front, 6 inches at sides and rear and 2 inches on top; the barbette, 6 inches front and 4 inches rear.

For the conning tower and shield the thickness is 9 inches throughout, with 2-inch top plates, and a 6-inch central tube having an internal diameter of 36 inches.

The protective deck will be of two courses, the lower one of 20-pound ship steel and the upper of 40-pound nickel-steel on the flat and 100-pound nickel-steel on the slopes and at ends.

A cofferdam about 30 inches wide will extend aft from the bow to within 16 feet of the stern.

The propelling machinery will consist of two vertical, inverted, 4-cylinder, direct-acting, triple-expansion engines, designed to develop 16,500 horsepower when running at 120 revolutions per minute with a steam pressure of 250 pounds in the high-pressure cylinder.

The cylinder diameters are 32½, 53 and (two) 61 inches, with a common piston stroke of 48 inches.

The boilers, of the Babcock & Wilcox type, twelve in number, have a total grate surface of 1,100 square feet, and 52,752 square feet of heating surface.

There are two main condensers, each with 10,375 square feet of cooling surface.

Provision is made for a complement of 856 officers and men.

Bennington.—*Boiler Explosion.*—On the morning of July 21st, while getting up steam, preparatory to proceeding to sea, the collapse of a furnace in one of the *Bennington's* boilers resulted in the death of one officer and sixty-one enlisted men, the injury, more or less seriously, of forty others, and much damage to the ship.

The lower furnace of boiler B (the port forward one) collapsed, causing this boiler to be driven aft against boiler D, liberating the steam contained in both boilers, wrecking the steam piping and, through breaking of sea valve connections, permitting entrance of water to hull.

The vessel began to settle with heavy list to starboard,

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making it necessary to beach her, with the assistance of tugs, to prevent sinking in deep water.

The heavy loss of life was caused principally by scalds and burns, due to escaping steam and water. No one in the fire-rooms escaped.

The accident is supposed to have been due to the accidental closing of the valve or pipe leading from boiler B to its steam gauge and the failure of the safety valve to lift upon the formation of an excessive steam pressure. The steam gauge, being disconnected, registered no pressure, and the water tender in charge was evidently not aware of the fact that pressure had formed until a few moments before the explosion, when a slight leak in the boiler was detected.

An examination and test of the material of the collapsed furnace showed it to be of good quality.

The boilers of the *Bennington* are about fourteen years old, having been in the ship since the time of her building in 1891; they are four in number, of the low, cylindrical, straightway, fire-tube type, with three furnaces each.

Wyoming—*Accident to.*—On the morning of July 18th, while at sea, *en route* from Panama to San Francisco, the U. S. S. *Wyoming*, lost her starboard propeller, the shaft having fractured just abaft the strut bearing. An examination by divers showed evidence of an old crack.

Paducah—*Trial of.*—The standardization and four-hour official trials of the gunboat *Paducah* were made at Newport, R. I., on August 15th and 16th.

The vessel, which was required by the contract to maintain an average speed of 12 knots for four consecutive hours, realized 12.85 knots without difficulty or mishap.

A complete description of this vessel and her sister ship, the *Dubuque*, appears in the first part of this number of the JOURNAL.

AUSTRIA.

Kaiman—*Launch of.*—The first-class torpedo boat *Kaiman*, built for the Austro-Hungarian Government by Messrs. Yarrow & Co., was successfully launched from their yard at Poplar

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on Friday, June 2d. She is the first of a special type, the authorities in Vienna having determined to completely reorganize their torpedo flotilla upon most modern lines. The *Kaiman* will be propelled by a single screw. The machinery will be similar to that adopted in the Austrian destroyer *Huszár*, launched on the 31st ultimo by the same constructors, the difference between the two types of vessel being that the destroyer will be fitted with two sets of machinery, while in the torpedo boat there will be one set, the machinery in the two classes being identical, the object of which is to secure interchangeability throughout the torpedo flotilla to the fullest possible extent. A large number of these vessels is being constructed in Austria, the *Huszár* and *Kaiman*, built by Messrs. Yarrow, serving as patterns. The design of both types will be generally similar to corresponding vessels built by the same firm for the Japanese Navy, subject to improvements which recent experience has indicated as desirable.

The Destroyer Huszár.—Last week a trial was made, off the mouth of the Thames, of the destroyer *Huszár*, built by Messrs. Yarrow & Co. for the Austro-Hungarian Government. This vessel, as already stated in a former issue, has been built to serve as a pattern vessel for a large number of sister craft which are to be constructed in Austria, the naval authorities of that country having determined to reconstruct their torpedo flotilla. To this end they have also ordered from Messrs. Yarrow & Co. a first-class torpedo boat to serve as a model for similar boats which are to be built in Austria. Of these craft, five destroyers are being constructed at the Stabilimento Technico, of Trieste, and six will be built at Fiume. Nineteen of the torpedo boats will be built at the former establishment and ten at Fiume. The *Kaiman*, which is the name of the model torpedo boat launched at Poplar on June 2d, is 180 feet 6 inches long and 18 feet 6 inches wide, the draught being 8 feet 6 inches and the displacement 200 tons. The propelling engines will develop 3,000 indicated horsepower, and the speed is to be 25¾ knots with a load of 55 tons. The armament will consist of four 1.85-inch guns and three 18-inch

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torpedo tubes. The vessel's complement will be two officers and twenty-five men.

To return to the destroyer tried last week, her length is 220 feet and her breadth 20 feet 6 inches, the depth being 13 feet. The total displacement with 100 tons on board—the contract trial weight—is 400 tons. The hull is constructed of the high-tensile steel which is made especially for torpedo craft. The conning tower is of chrome-steel and is bullet proof at a range of 20 yards point blank. The armament consists of two 18-inch torpedo tubes on deck, one 76-millimeter gun on a platform forward, and seven 47-millimeter quick-firing guns. Provision is made for carrying two spare torpedoes.

The propelling machinery consists of two sets of four-cylinder three-stage compound engines, driving twin three-bladed screws, each set developing 3,000 indicated horsepower, or 6,000 indicated horsepower collectively. The four-crank engine has been preferred, although it is heavier and more costly, because it gives less vibration than the three-cylinder arrangement. The engines are, of course, balanced on the Yarrow-Schlick-Tweedy system, and on the trial the *Huszár* ran at full speed in a most satisfactory manner in regard to vibration. Vibration, it may be added, is not only a question of convenience and comfort, but it has been found to be a matter that affects the health of the crew after a period of continuous running.

There are the usual auxiliaries, including electric-light installation for lighting the vessel and for the projectors, steam-steering gear and evaporating and distilling apparatus. There is a steam capstan forward. There are two 8-cwt. anchors of the ordinary Admiralty pattern. In order to provide for the housing of these, without masking the fire of the forward guns, a special tilting table has been devised by the Austrian authorities. The boilers are four in number, and are of the Yarrow straight-tube variety. The destroyer will have a complement of five officers and sixty-four men.

The *Huszár* is interesting, as she is similar—excepting in some details suggested by recent experience—to the destroyers

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which Messrs. Yarrow & Co. built some short time ago for the Japanese Government, and which have been doing such excellent service for their side during what may, perhaps, be described as the late naval war. The Japanese boats, however, made a speed of 31 knots on trial, whilst the contract speed of the *Huszár* is 28 knots; but the Japanese boats were tried with a load of 40 tons, whilst, as stated, the trial load of the Austrian vessels is to be 100 tons. This is a reduction of speed of 1 knot for about every twenty tons additional weight.

The most marked difference between the Austrian and Japanese boats consists in the arrangement of the funnels and uptakes of the boilers. With these high-powered vessels the boiler space necessarily occupies a considerable length of the forward middle part of the hull, and this brings the foremost chimney very near the bow. This in turn pushes forward the navigating bridge, so that there is quite a short length of the vessel in front of the steering wheel. Every sailor knows how much easier it is to maneuver a vessel when the whole length is in front of him, but this cannot be arranged with modern steamers of this type, and, practically, the bridge must be in front of the funnels. With three-quarters or more of the ship out of sight, and that three-quarters swinging through the largest arc when the helm is put over, it is extremely difficult to maneuver these light craft in narrow or crowded waters, especially when a strong wind is blowing, whilst the tactical advantage of accurate steering can hardly be over estimated.

To get the boilers into the vessel and to keep the bridge a reasonable distance from the bow was the problem to be solved, and this has been done in the *Huszár* in a very simple manner. The chimney of the forward boiler—there is a funnel to each boiler—is carried aft by bringing the uptake horizontally over the stokehold by means of a raised superstructure, and in this way an addition of 17 feet or 18 feet of "sighting" is secured. There appears to be no serious obstacle to the plan being still further extended, so that the wheel could be placed still further aft, or the bridge being kept further away from the chimneys—a desirable feature with a leading wind.

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The trial of last Thursday week was of a preliminary nature in regard to speed, the time being chiefly devoted to the usual anchor trials. Four runs were, however, made on the Maplin mile, the vessel realizing the contract speed of 28 knots with the boilers steaming easily, the pressure averaging about 200 pounds to the square inch, or 65 pounds less than that at which they are designed to work, and the revolutions averaging 380 per minute.

The official full-speed trials took place on Tuesday, June 20, when a speed of 28.537 knots was attained during a continuous run of three hours, the vessel being loaded with 100 tons. The Austro-Hungarian Government was represented by Captain Josef Ritter von Schwartz, naval attaché; Captain Franz Ritter von Keil; Baron Franz von Preuschen; Marine Engineer Rudolf Zhernotta; Herr Josef Grond; Engineer Anton Tonsa; Herr Hans Musurka, and Herr J. Seifridsberger. The vessels were constructed under the two last-named officials. Mr. W. W. Marriner and Mr. Harold Yarrow were in charge of the trials on behalf of Messrs. Yarrow & Co.—"Engineering," London, June 23.

BRAZIL.

The additions which Brazil proposes to make to her fleet are expected to comprise three iron clads, with a displacement of 12,500 tons to 13,000 tons each; three armor-plated cruisers, with a displacement of 9,200 tons to 9,500 tons each; six torpedo-boat destroyers, with a displacement of 400 tons each; six torpedo boats with a displacement of 50 tons each; three Holland submarines, and a transport with a displacement of 6,000 tons. It is proposed that the iron clads shall each carry sixteen guns; the armor-plated cruisers are to steam at the rate of 21 knots, and the torpedo-boat destroyers are to attain a speed of 31 knots. The execution of the program is to extend over nine years, divided into three terms of three years each. The cost which it will involve is estimated at £7,120,000.—"Engineering," London.

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ENGLAND.

Hibernia—*Launch of.*—The battleship *Hibernia*, of the *King Edward VII* class, was launched at the Devonport dockyard on June 17th. The principal dimensions are: Length between perpendiculars, 425 feet; extreme breadth, 78 feet; draught of water forward, 26 feet 3 inches; aft, 27 feet 3 inches, and load displacement, 16,350 tons. The engines will consist of two independent sets of the vertical triple-expansion type, each having four cylinders. Each set will be capable of developing 9,000 indicated horsepower, with 120 revolutions per minute. The speed contracted for at full power is 18½ knots. There will be eighteen water-tube boilers of the Babcock & Wilcox type, and three cylindrical boilers. The armament consists of four 12-inch breech-loading guns, worked in two barbettes, four 9.2-inch breech-loading guns in four barbettes, ten 6-inch breech-loading guns worked in a battery, twelve 12-pounder, 18-cwt., quick-firing guns, two 12-pounder, 8-cwt., quick-firing guns, fourteen 3-pounder quick-firing guns, two .303 Maxim guns and four submerged torpedo tubes. The total complement of officers and men will be 781.

(For further data as to this type, see Vol. XVI, pp. 998 and 1315, and Vol. XVII, p. 134.)

Achilles.—This British first-class armored cruiser was launched on June 17 from the yard of Armstrong, Whitworth & Co., at Elswick. The *Achilles* is one of four vessels provided for in the naval estimates of 1903–4. These vessels are known as the Warrior class, built on French lines, with guns in single turrets placed high up. The *Achilles* is 480 feet long, 73 feet 6 inches broad, 40 feet 1½ inches deep, with a draught of 27 feet. Her displacement is 13,660 tons, and her armament is as follows: Six 9.2 and four 7.5 breechloading guns, two 12-pounders and twenty-four 3-pounder quick-firing guns, two Maxim guns and three submerged torpedo tubes. The armor belt is six inches thick. Her speed will be 22 knots an hour.

(For further details see Vol. XVII, p. 546.)

Roxburgh—*Trial of.*—This cruiser, built by the London and Glasgow Shipbuilding Co., Glasgow, for the Admiralty, pro-

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ceeded to sea on her official trial trip recently. She is the fourth of the cruisers of the *Devonshire* class which have passed through their steam and gun trials, and her speed record, as the mean of four runs over the measured mile on the Clyde, was 23.63 knots (the highest attained by any vessel of the class), to attain which the *Roxburgh* developed more power than the others. The engines throughout the eight-hours' trial maintained an average of 21,857 indicated horsepower, which is 857 indicated horsepower over that guaranteed in the contract. The starboard engine made 143 revolutions per minute, and developed 10,734 indicated horsepower, whilst the port engine for 144 revolutions indicated 11,123 indicated horsepower. The air pressure in connection with the tank boilers was 1⅝ inches, and in the water-tube boilers 1 inch, the latter of the Durr type. The steam pressure was 204 pounds at the boilers, and the coal consumption worked out to 2.31 pounds per indicated horsepower per hour. In the first of the coal-consumption trials of thirty hours' duration, to Portsmouth, with the engines making 86½ revolutions per minute, the power averaged 4,634 indicated horsepower, giving the ship a speed of 14.4 knots. The coal consumption on this trial was at the rate of 2.09 pounds per indicated horsepower per hour. The gun trials having been carried out in the English Channel, the *Roxburgh* returned to the Clyde, making the second of the coal-consumption trials of thirty hours' duration, when, at an average rate of 127.4 revolutions, 15,005 indicated horsepower was shown, and the coal consumption 1.991 pounds per indicated horsepower per hour. On four runs over the measured mile at Skelmorlie, the mean speed was 21.54 knots.

(For detailed description of this class see Vol. XVII, pp. 430 and 443.)

Scout Skirmisher—*Trials of.*—H. M. S. *Skirmisher*, the second of the vessels of the class designed and constructed complete for commission by Messrs. Vickers Sons & Maxim, Limited, at the Naval Construction Works, Barrow-in-Furness, completed this week her specified trials, which, as in the case

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of the other vessels, included a 96-hours' coal-consumption test, an 8-hours' full-power trial, as well as gun, circling and general maneuvering performances. The first of the vessels was the *Sentinel* (see JOURNAL A. S. N. E., Vol. XVII, page 208). She has now been in commission for some time, is now engaged on an extended cruise, and, peculiarly enough, was at anchor at the Tail of the Bank on the Clyde, to greet her sister ship on her return from the full-power test on June 6th. Although we have already described the vessels of the class, it may be said here that the Vickers ships are 360 feet in length, 40 feet in beam, and at their full load displacement of 2,940 tons they carry a coal supply sufficient to enable them to steam at a cruising speed of between 10 and 12 knots for 1,500 miles. The draught under those conditions is 14 feet 3 inches. The constructors guarantee a speed at load draught of 25 knots, and, to attain this, triple-expansion engines, driving twin screws, were fitted to develop 17,000 indicated horsepower. The *Sentinel* on her trials easily exceeded this guarantee, and now the *Skirmisher* has repeated the same satisfactory performance, her mean speed on the 8-hours' run being 25.2 knots, when developing 16,899 indicated horsepower.

The trials of the *Skirmisher* have been carried out within a comparatively short period of time. She only left Barrow on May 23, and after a preliminary trial in Morecambe Bay proceeded to the Clyde, where she arrived on May 24th. The first preliminary progressive trial was carried out on the 25th, when the vessel worked up to a speed of 20 knots. Two days later she continued the progressive runs, the machinery developing the necessary power to give the speed of over 25 knots. The vessel then proceeded on her 96-hours' coal-consumption trial, beginning on May 30 and terminating on June 3d. The performance on this trial showed the machinery to be economical. On the first half of the trial the speed attained was 10.6 knots for 904 indicated horsepower, and the coal consumption showed that 10.3 nautical miles were covered for each ton of coal consumed. On the second half of the trial

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the results were equally satisfactory; the speed was 10.9 knots for 995 indicated horsepower, and the coal consumption was equal to 1 ton for each 10 nautical miles steamed. Table I gives the official results for the 96-hours' trial. With a mean steam pressure in the boilers of 177 pounds and at the high-pressure receiver of the engines of 44 pounds, the power for 79.6 revolutions was 949 indicated horsepower. This gave a speed of 10¾ knots, while the coal consumption worked out to practically one ton per 10 knots steamed. The coal consumption is debited with the cost of making up the loss of water in the evaporators; although only 15.9 tons were lost during the four days and nights of steaming, the evaporators made 16.67 tons. One more point which ought to be noticed is that the vessel had not been in drydock since April 17; and when later she was docked, preparatory to the 8-hours' full-power trial, it was found that the hull was by no means clean.

Table I.—RESULTS OF 96-HOURS' COAL-CONSUMPTION TRIALS OF H. M. S. *SKIRMISHER*, MAY 30 TO JUNE 3, AT CRUISING SPEED.

—	Starboard.	Port.
Mean steam in boilers, pounds	177	177
at engines (high-pressure receiver), pounds	47	41
air pressure in stokeholds	nil.	nil.
cut off in high-pressure cylinder, per cent	35	35
vacuum, inches	25.7	25.5
revolutions per minute	79.6	79.7
Mean pressure in cylinders:		
High, pounds	20.9	18.9
Intermediate, pounds	5.62	4.78
Low, forward, pounds	1.7	2.32
Low, aft, pounds	1.63	2.35
Indicated horsepower:		
High	210	189
Intermediate	144	123
Low, forward	60	83
Low, aft	57	83
Total indicated horsepower	471	478
Gross total indicated horsepower	949	
Coal, per indicated horsepower, pounds	2.5	
Sea miles per ton of coal	10.075	
Speed, knots	10.756	
Loss of water (tons per 1,000 indicated horsepower per 24 hours)	4.21	
Total loss of water for all 96 hours, tons	15.9	
Water made by evaporators, tons	16.67	

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The eight-hours' full-speed trial was run on June 6th, and the weather conditions were satisfactory, there being only a slight breeze from the northeast. The vessel, soon after leaving the anchorage, commenced her trial at 9 o'clock, and six runs were made over the measured mile, occupying 1 hour and 25 minutes. The vessel thereupon continued down the Firth, and the speed for the whole eight hours was measured by the average number of revolutions required on the six runs over the measured mile for 25 knots. The revolutions for 25 knots are almost exactly 200, and throughout the succeeding 6½-hours' run the engines made 201.55 revolutions, the power being on the 6½-hours' run 17,013 indicated horsepower, while for the whole of the eight hours the mean was 16,899 indicated horsepower. The average speed was 25.2 knots. We append the mean results attained during the 6½ hours following the measured-mile performance:

Table II.—RESULTS OF FULL-POWER TRIAL OF H. M. S. *SKIRMISHER*, JUNE 6, 1905.

—	Starboard.	Port.
Mean steam in boilers, pounds	253	253
at engines (high-pressure receiver), pounds	239	239
Air pressure in stokeholds, inches	2.3	2.3
cut off in high-pressure cylinder, per cent	75	75
vacuum, inches	25.5	25.3
revolutions per minute	201.5	201.6
Mean pressure in cylinders:		
High, pounds	103.5	106.1
Intermediate, pounds	36.7	38.8
Low, forward, pounds	19.7	16.4
Low, aft, pounds	19	19.6
Indicated horsepower:		
High	2,624	2,693
Intermediate	2,382	2,520
Low, forward	1,757	1,465
Low, aft	1,818	1,754
Total indicated horsepower	8,581	8,432
Gross total indicated horsepower	17,013	
Water loss per 1,000 indicated horsepower per 24 hours, tons	2.33	
Total loss of water for all 8 hours, tons	13.16	

These results are very satisfactory. The starboard and port engines ran at almost the same speed, and gave practically the

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same indicated horsepower. The trial concluded at 5 o'clock, and the ship anchored. On the completion of the gun, anchoring and circle trials on Thursday, the vessel returned to Barrow to be completed for commission. Messrs. Vickers Sons & Maxim, Limited, will be the first firm to deliver their pair of scouts.—"Engineering," London.

Scouts Pathfinder and Patrol—*Trials of.*—The length between perpendiculars is 370 feet; breadth, 38½ feet; and the vessels have a displacement at normal draught of 2,850 tons. The hull is built generally of mild steel, with special steel in parts of the structure subjected to the greatest stresses, and is well subdivided into numerous watertight compartments. A steel protective deck worked in the region of the water line (except over engines, where it is at the level of the upper deck,) extends from stem to stern, being ⅝-inch thick on the flat portions and 1½ inches thick on the slopes. Side armor of 2-inch K. N. C. steel is worked in the wake of the engines, extending from well below the water line to the height of the upper deck—an arrangement which conduces to large and airy engine rooms.

The *Pathfinder* encountered very rough weather on her earlier official trials, and won high praise for her steadiness in a seaway. The coal stowage in these two vessels was made a special feature of the design, and bunkers are provided both above and below the protective deck, with a capacity sufficient to give the vessel a radius of action of 6,000 miles at an economical speed.

The vessels all carry an armament of ten 12-pounder quick-firing guns of the new 18-cwt. pattern, eight 3-pounder quick-firing guns, and two 18 inch torpedo-tubes. Three of the 12-pounders are placed on the forecastle, three on the upper deck aft, and the remainder in the waist of the vessel. Cabin, messing and sleeping accommodation is provided for a complement of 286 officers and men.

The machinery consists of two sets of four-cylinder triple-expansion engines in separate watertight compartments, balanced on the Yarrow-Schlick-Tweedy system, the low-pres-

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sure cylinders being placed at the ends. To obtain the full advantage of the keelsons for supporting the machinery, and to maintain the longitudinal keelsons as deep as possible, the engine-bed plates are made of special design, sunk in and secured between the longitudinals, thus securing a very strong and stiff arrangement with a minimum of weight. The valve gear is of the ordinary double-eccentric double-bar link type. Piston valves, taking steam inside, are fitted to the high and intermediate engines, the low-pressure being fitted with double-ported flat slide valves having relief rings at the back. The cylinders are of the following dimensions: High-pressure, 32½ inches; intermediate pressure, 51½ inches; and each low pressure, 58 inches in diameter, with a stroke of 2 feet 6 inches. The steel shafting is hollow throughout, the crank shaft being 13½ inches external diameter, with 6⅝ inches bore. The condenser shells are of sheet brass, 6 feet 2 inches in diameter, one for each engine, the total cooling surface being 14,000 square feet. Special arrangements have been made to ensure a minimum of vibration to the tubes. The air pumps are Weir's independent type, one double pump being fitted to each engine. The pipes to the centrifugal circulating pumps are 17 inches in diameter. Special attention has been given in the design to the lubricating arrangements on the main engines, the outcome of which has been satisfactory. There are twelve water-tube boilers of the well-known Laird type, fired from four stokeholes. The working pressure is 275 pounds per square inch. The total heating surface is 43,800 square feet, and the total grate area 656 square feet.

The contract trials for each boat consisted of a 96-hours' trial at between 10 and 12 knots and a full-power trial of eight hours, during which a speed of 25 knots had to be obtained. Throughout these trials the machinery worked in the most satisfactory manner, and in both vessels the trials were completed within a very short period. H. M. S. *Pathfinder* commenced her 96-hours' trial on March 26, and ran her full power on April 8. H. M. S. *Patrol* ran her 96-hours'

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on June 1 to 5, and her full-power trial on June 17. The following are some of the results, the full-power records being those for 6½ hours after completing the miles.

	H. M. S. *Pathfinder.*		H. M. S. *Patrol.*	
	96 hours.	Full power.	96 hours.	Full power.
Speed (knots)	10.988	25.345	10.969	25.062
Revolutions	85.4	220.2	84.7	213.5
I.H.P.	1,063	17,235	1,164	16,433
Coal per I.H.P., pounds	2.35	...	2.18	...
Water loss per 1,000 I.H.P., per 24 hours, tons	1.82	3.13	1.71	1.33

All steering and maneuvering trials were successfully carried out, and both vessels are rapidly approaching completion for commission; in fact, the *Pathfinder* will leave Birkenhead during the first week of July, well within her contract date.—"Engineering," London.

Steam Trials of H. M. S. Foresight—The scout *Foresight*—the second of the vessels of this type, built by the Fairfield Shipbuilding and Engineering Company—has concluded her official trials with most satisfactory results. This vessel is in all respects the same as the *Forward*, which completed her trials in May last; and as we then entered into details of the design of the ship and her machinery, we need not refer further to this matter than to say that the length of the vessel between perpendiculars is 365 feet and the beam 38 feet 9 inches. The designed displacement is 2,945 tons when the vessel has a draught of 14 feet 5 inches. The machinery is of the four-cylinder triple-expansion type, designed to develop 16,500 horsepower; but, as will presently be shown, the *Foresight*, like her sister ship, the *Forward*, exceeded the required speed, the power being only 14,300 horsepower, while in the *Forward* it was 15,000 horsepower. The boilers are of the modified Thornycroft type, with 42,960 square feet of tube surface and 752.4 square feet of grate area.

The first trial consisted in a coal-consumption test of 98 hours' duration at an economical cruising speed, and, as in

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the case of the *Forward*, 10½ knots was found to be the most satisfactory rate. This speed, as is shown in Table I, was realized for 810 horsepower; the rate of coal consumption being 2.5 pounds, which is an improvement on the *Forward*. In that vessel the power was 29 indicated horsepower greater and the coal consumption 0.16 pound per unit higher.

Table I.—RESULTS OF 96-HOURS' COAL-CONSUMPTION TRIAL OF H. M. S. *FORESIGHT*, JUNE 24, 25, 26 AND 27, 1905, AT 10.552 KNOTS SPEED.

Draught, forward, 13 feet, 2½ inches; aft, 14 feet, 10½ inches.

—	Starboard.	Port.
Mean steam in boilers, pounds	167	167
at engines (high-pressure receiver), pounds	50	53
air pressure in stokeholds	nil.	nil.
cut-off in high-pressure cylinder, per cent	33	33
vacuum, inches	25	23.8
revolutions per minute	80.47	80.16
Pressure in cylinders:		
High, pounds	15.75	17.4
Intermediate, pounds	4.55	5.15
Low, forward, pounds	2.17	1.85
aft, pounds	2.17	1.9
Indicated horsepower:		
High	151	166
Intermediate	111	125
Low, forward	70	58
aft	70	59
Total indicated horsepower	402	408
Gross total indicated horsepower	810	
Coal, per indicated horsepower, per hour, pounds	2.5	
Water loss per 1,000 indicated horsepower, per 24 hours, tons	3.56	
Indicated horsepower on mile	829	
Speed, mean of 24 runs on mile, knots	10.697	
96 hours, knots	10.552	

As a consequence of the reduced consumption the coal to be carried on the full-power trial was considerably less, so that the displacement was much below that designed for. Loaded with coal to enable her to run at 10½ knots for 1,500 miles at the rate of coal consumption and the power mentioned, the vessel proceeded on June 30 on her full-power test, which was to consist of six runs over the measured mile at Skelmorlie, steaming for 6½ hours afterwards, making 8 hours in all, at a guaranteed rate of 25 knots. This result was easily realized, the air pressure in the stokeholds throughout the trials being between 1 inch to 1½ inches, while the cut-off

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at the high-pressure engines was at 60 per cent. of the stroke. As shown in Table II, the engines worked at practically 211 revolutions, and averaged 14,330 indicated horsepower, the mean speed being 25.224 knots.

Table II.—H. M. S. *FORESIGHT*. SIX RUNS ON MEASURED MILE AT FULL SPEED, JUNE 30, 1905.

Run.	Revolutions.	Speed.	Indicated horsepower.
		knots.	
1	206.7	25.175	13,723
2	214	25.14	14,956
3	211.2	25.46	14,732
4	211.2	25.035	14,770
5	208 7	25.316	14,029
6	207.5	24.76	13,772
Mean	210.99	25.224	14,330

This is slightly better than in the case of the *Forward*, which for 15,000 horsepower gave 25.286 knots. The difference, however, is probably accounted for by the less amount

Table III.—RESULTS OF FULL-POWER EIGHT-HOURS' TRIAL OF H. M. S. *FORESIGHT*, JUNE 30, 1905.

—	Mean of six runs on measured mile.		Mean of subsequent 6½ hours of trial.	
	Starboard.	*Port.*	*Starboard.*	*Port.*
Mean steam in boilers, pounds	245	245	245	245
Mean steam at engines (high-pressure receiver), pounds	230.5	229	232.8	232
Mean air-pressure in stokeholds, inches	1	1	1.4	1.4
Mean cut-off in high-pressure cylinder, per cent	60	60	60	60
Mean vacuum, inches	25	25	25	24.7
Mean revolutions per minute	210.1	209.6	210.3	210
Mean pressure in cylinders:				
High, pounds	107.6	110	107.4	110
Intermediate, pounds	33.4	33.4	32.6	32.5
Low, forward, pounds	14.4	14	14.3	14.5
aft, pounds	14.3	14.6	14.5	15
Indicated horsepower, high	2,673	2,734	2,669	2,731
intermediate	2,131	2,129	2,114	2,069
low, forward	1,175	1,136	1,167	1,169
aft	1,161	1,191	1,179	1,210
Total indicated horsepower	7,140	7,190	7,129	7,179
Gross total indicated horsepower	14,330		14,308	
Total loss of water for all 8 hours, tons	5.17			
Water loss per 1,000 I.H.P. per 24 hours, tons	1.08			
Speed, knots	25.224		25.123	

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of coal required to be carried in the bunkers, so that when the bunkers are carrying their full load the vessel will easily exceed her speed while giving a much greater radius of action. In Table III the result of the engine's performance, not only on the six runs over the measured mile, but on the subsequent 6½-hours' trial, are set out.

Throughout the whole six hours the vessel steamed very easily, while the loss of water was only about 1 ton per 1,000 horsepower per 24 hours, as compared with 2.71 tons in the case of the *Forward*.—"Engineering," London.

Dreadnought.—The new British battleship *Dreadnought*, included in the navy program of the current financial year, is to excel any vessel yet built, and for this purpose will be fitted with ten 12-inch guns. Hitherto no ship has had more than four of these large weapons, and although this latest vessel will not carry any other pieces except the machine guns for warding off torpedo attack, her fighting power will be far in excess of that of her predecessors. Prior to the vessels of the *King Edward* class the usual armament for British battleships was four 12-inch guns and ten or twelve 6-inch quick-firers, the collective muzzle energy of the sixteen guns being about 210,000 foot tons. In the *King Edward* class an improvement was made by adopting four 9.2-inch guns in addition to four 12-inch and ten 6-inch weapons. This increased the muzzle energy of the guns collectively to about 275,000 foot tons. In the *Lord Nelson* class the 6-inch weapons were eliminated and four 12-inch and ten 9.2-inch adopted. At the same time it was decided to increase the length and power of each weapon, so that the total muzzle energy from one round from all the guns became 433,000 foot tons. Now the 9.2-inch gun is discarded and only 12-inch weapons installed, and the ten of these together will give an energy of 480,000 foot tons. This change, it is pointed out, simplifies the arrangements for ammunition on board and at naval bases, so that after action a ship can easily get a new supply. The battleships will be of 18,000 tons displacement, and fitted with turbine machinery and Babcock & Wilcox boilers of 23,000

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indicated horsepower, giving a speed of about 21 knots, as compared with 19 knots for the ships of the *King Edward* class.—"Army and Navy Register."

Destroyers.—Contracts have now been arranged for the construction of five sea-going destroyers included in the navy program for the current year, and these vessels will be built, one each by Messrs. J. I. Thornycroft, Limited, Chiswick; Sir W. G. Armstrong, Whitworth & Co., Limited, Elswick; Messrs. Cammell, Laird & Co., Limited, Birkenhead; Messrs. R. and W. Hawthorn, Leslie & Co., Limited, Newcastle-on-Tyne; and Messrs. J. S. White & Co., Cowes. These vessels are a great advance upon anything that has yet been done in torpedo-boat destroyers, as the speed guaranteed, under penalty, by the contracting firms is to be 33 knots when the vessels are carrying a load of coal sufficient to give them a radius of action of 3,000 miles at about 10 knots speed. This condition is much more severe than was exacted in the case of the 30-knot torpedo boats; and, moreover, the scantlings proposed for these high-speed craft will be considerably heavier. It is true that speeds of 32 to 33 knots were attempted by one or two builders in connection with the 1897–8 program, but the highest realized on trial was 31½ knots. Even then the boats were not required to meet the same stringent conditions as is the case with these new 33-knot craft. The boilers are to be of the Express type, adaptable for using oil fuel, and the propelling machinery in all cases will be of the Parsons turbine type, with arrangements similar to those in the *Amethyst*, where special turbines were fitted for low-speed cruising. The displacements of the boats will average over 800 tons, so that very considerable power will require to be developed to attain the speed. It may be noted that the 30-knot destroyers were from 300 to 360 tons displacement. When the scantlings were increased for what is now known as the "River" class, the displacement was increased from 525 to 550 tons; but the speed was reduced to 25½ knots, under the same fuel-capacity conditions as in the new boats. It will therefore be seen that the new vessels will mark a great step in speed, and

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the contracts carry with them corresponding responsibilities. It is the intention to order later an experimental destroyer, which it is expected will have a speed of 36 knots.—"Engineering," London.

Submarine A8—*Accident to.*—Another deplorable disaster to a submarine occurred at Plymouth on June 8th, when *A8*, while at exercise for instructional purposes off the breakwater, foundered without giving any warning to her unfortunate crew. The conning towers were open, when suddenly the bow of the vessel dipped heavily, which caused her to ship a large quantity of water through the conning tower, and before it was possible to close the conning tower she went down. Lieutenant Candy (the commander), Sub-Lieutenant Murdock, and two petty officers, who were on deck at the time, were saved; but Sub-Lieutenant Fletcher and nine of the crew, who were below, were all drowned. Steps were immediately taken to raise the vessel, which was successfully accomplished on the morning of June 12th. The finding of the courtmartial which was held on the survivors was as follows:

"The Court finds that submarine *A8* was lost outside Plymouth breakwater, about 10.30 A. M., on the 8th June, through foundering from water getting in through the open conning tower when running at ten knots speed with a buoyancy at starting of six tons and trimmed four degrees by the stern, which buoyancy and trim appeared to have altered from some unexplained cause, such as the movement of weights forward, possibly admission of water, or free water. There is no evidence to show which way the diving rudders were put, although repeated orders were given to put them up. They may, however, have been put inadvertently the opposite way to that actually intended. Water having once commenced to get into the conning tower and running forward, she could not recover herself, although the helm was finally found hard up.

"The Court regrets that the officer in command did not stop the engines sooner when he realized that his repeated orders

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to put the diving rudders up had no effect, although expert evidence shows there was no reason to anticipate danger in running the boat trimmed as at starting, as it appears to have been considered impossible for her to dive under these conditions. The Court, therefore, acquits Lieutenant Algernon Hugh Chester Candy and the other survivors of submarine *A8* of all blame for the loss of that vessel."

The accident to *A8* is the third to British submarines which has been attended with fatal results. In March of last year *A1* was run down at Spithead, with a loss of eleven lives. In February of this year there was an explosion of petrol vapor on board *A5* at Queenstown, by which four men met their death.—"J. R. U. S. I."

Magnificent—*Accident on Board.*—A serious accident occurred on board the *Magnificent* on June 14th, when at target practice off Tetuan, on the Moorish coast, by which an officer and four men lost their lives and twelve others were more or less injured. It appears there was a miss-fire in one of the 6-inch guns; the breech of the gun was opened to ascertain the cause, when the cartridge exploded, igniting two others in the casemate.

FRANCE.

Submarines.—In view of the fact, as announced in these columns recently, that the French naval authorities have recently launched one of the largest submarine boats in the world, her displacement being 213 tons, and her engines having 250 horsepower, it is interesting to observe that the French are also evolving submarines of so small a type that they can be carried on battleships and cruisers, to be placed in the water only when needed. One boat of this type, the *Volta*, now under construction, is said to be only 36 feet long. The maximum diameter of her circular cross sections, according to the "London Times," is 6 feet 9 inches, and her displacement is less than 20 tons. We learn from the same authority that she will use electrical power alone, and that her estimated speed limit is about eight miles an hour. She is to have a crew of three men only and to carry two torpedoes, one slung on each

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side with suitable detaching gear. It is stated that the preliminary trials have been successful, and it is to be hoped that further information concerning this novel vessel may soon be forthcoming. This new type of submarine is being constructed by private builders under government supervision, and the experiment is a matter of keen interest to submarine experts in Europe. This is the first time a private company has undertaken to build a submarine in France, and the attitude of the French Admiralty is watched with great interest by all concerned, as they have not themselves undertaken the construction of similar classes of submarines, but have pursued a policy of continuous development in size and cost.—"Army and Navy Journal."

Destroyers.—Commenting on the prominence to be given to destroyers in the naval program of France, as fifteen of this type of vessel are to be put under construction in the French shipyards this year, a much larger number than has been laid down at one time in any previous year, the "London Engineer" says that the introduction of the scout has induced the French naval department to turn its attention to the necessity of efficiently armoring the new destroyers so as to protect the engines and boilers from light artillery at any range, while they will be invulnerable to the fire of guns of 76 mm. bore at distance from 2,000 m. to 3,000 m. The danger of a destroyer being put out of action or sunk by quick-firing guns has been so far increased by the superior armament and higher speed of the scout that the destroyer can no longer hope for safety in flight, and in the opinion apparently of M. Normand, who is responsible for the designs of the destroyers, the efficiency of these craft can only be maintained by protecting the vital parts against such fire as may be brought to bear upon them by the scout. In the new destroyers the displacement will be increased from 330 to 450 or, perhaps, 470 tons, so that the engines, boilers and other vital parts may be protected by armor plate 40 mm. to 50 mm. in thickness. The speed will be two knots less than that of the 330-ton craft, but they are expected

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to be more seaworthy in heavy seas.—"Army and Navy Journal."

Farfadet—*Accident to.*—The French submarine *Farfadet* met with an accident, July 6, at the entrance to the port of Sidi Abdallah, Tunis, under the following circumstances: Commandant Ratier, at the moment of plunging, noticed that one of the doors was not closed properly and rushed to try to adjust it hermetically. He was too late, the water rushing in with force and throwing out the commandant and two men, the boat sinking in ten meters and lying in the mud at the bottom. Twelve men were encased in the boat, and powerful tugs and a strong contingent of engineers have been working for a week, but without success, to rescue the men and bring up the boat. The entombed men, when divers went down on July 7, were reported alive in the after part of the submarine, and rapped on the inside of the boat to communicate with the divers. On July 7 the *Farfadet* was raised above the surface of the water long enough to ascertain that the occupants were alive, and fresh air was pumped into the vessel; but when an attempt was made to tow her into shallow water she sank again. The boat was raised again on July 9, and the imprisoned men were reported still alive, but almost on the point of suffocation. When the hatch had been got about a yard out of the water the crane broke and the boat sank again. A second attempt failed to raise the boat, and it was later reported that the men in her no longer responded to signals. The *Farfadet*, which is 125 feet long, has electric propulsion, four torpedo tubes, and has reversible propellers. Her air tanks and cabin accommodations are exceptionally good for a submarine boat, and this is attributed as the reason the prisoners were able to keep alive for so long a time.—"Army and Navy Journal."

GERMANY.

New Ships.—An iron clad, officially known at present as *Q*, and a division of torpedo boats, officially known as *G132*, *G133*, *G134*, *G135*, *G136* and *G137*, have been ordered to be built in the Germania yards at Kiel. An iron clad, offi-

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cially known at present as *R*, is to be built in the Schichau yards at Dantzic; an armor-plated cruiser, officially known as *D*, in the Blohm and Voss yards at Hamburg; and a small cruiser, officially known as *P*, in the navy yards at Dantzic.—"Engineering."

S124—*Accident to.*—The German battleship *Woerth* rammed torpedo-boat destroyer *S124* in the harbor of Eckernfoerde, July 5, cutting her completely in two, and three firemen in the boiler room, it is reported, were scalded to death by escaping steam. The *S124* was one of six torpedo-boat destroyers launched in 1904. She was of 420 tons displacement, and credited with a speed of 30 miles an hour.

River Gunboat.—A new river gunboat, which is to be sent out to Hong Kong in sections, has been constructed by the Schichau firm, Elbing. Her dimensions are as follows: Length, 157 feet 6 inches; beam, 26 feet 4 inches; draught, 2 feet, with a displacement of 168 tons. Her two engines will develop together 1,300 I.H.P., giving her a speed of 13 knots. Steam will be provided by two Thornycroft water-tube boilers. Her bunkers will hold 80 tons. She has a military mast, and is armed with one 3.46-inch 30-caliber quick-firing gun, one 1.97-inch 40-caliber quick-firing gun, and three 0.315-inch machine guns. Her complement will be fifty-three officers and men. Her deck house and sides are protected by a light armor of nickel-steel 0.315-inch in thickness.—"Neue Preussische Kreuz-Zeitung."

S125—*Trial of.*—The trial trip of the torpedo boat *S125*, built at the Schichau Works, Germany, and fitted with turbine engines, took place off Elbing, June 14. A dispatch states that she averaged 27.8 knots per hour for three hours. Her highest speed rate was 29½ knots per hour. Four sister boats, fitted with piston engines, averaged two knots per hour less than the *S125*.

JAPAN.

Katori—*Launch of.*—The battle ship *Katori* was successfully launched from the works of Messrs. Vickers Sons & Maxim on July 4th. A brief description of the vessel follows.

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The length between perpendiculars is 420 feet, and over all 455 feet 9 inches; the breadth is 78 feet, and the depth to upper deck, 44 feet. The draught in fighting condition will be 27 feet, when the displacement will be 15,950 tons. As to gun fire—which is the most important element in a ship of the line—the *Katori* will mount pairs of 12-inch guns in barbettes at the forward and aft end of the upper deck. These are of the Vickers type, weighing 57 tons, and they will deliver their 850-pound projectiles with a velocity and energy which will enable them at six miles range to perforate armor 9 inches in thickness, and at four miles range to defeat modern armor about 13 inches in thickness. There are four 10-inch guns, mounted singly in barbettes at each corner of the main citadel. The Vickers guns of this type fire a 500-pound shot with an energy sufficient to perforate at six miles range armor 6½ inches thick, and at four miles range, 9 inches thick. In the armament of the British vessels of the *King Edward VII* class the guns corresponding to these 10-inch weapons are only of 9.2 inches caliber, so that the Japanese ship has a considerable superiority. In addition, the *Katori* mounts twelve 6-inch guns within a concentrated casemate amidships; each gun, however, is entirely separated by armor screens.

This completes the main armament of the *Katori*. One round from these twenty main guns represents a collective energy of nearly 375,000 foot tons. As the 12-inch guns may fire two projectiles per minute, the 10-inch guns three rounds, and each 6-inch gun ten rounds, the total weight of shot which may be discharged in a minute is 24,800 pounds, representing a collective energy of approximately 1,400,000 foot tons. All four 12-inch guns may be trained on the broadside. The two 10-inch guns and six 6-inch guns are similarly utilisable on the broadside, so that one minute's possible broadside fire is 15,800 pounds, with a collective energy of over 900,000 foot tons. As two 12-inch guns, two 10-inch guns and two 6-inch guns may be fired ahead, the vessel may maintain a very hot pursuit, discharging per minute four projectiles of 850 pounds, six of 500 pounds, and twenty of 100 pounds.

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For the repelling of torpedo attack there are mounted, in protected positions in various parts of the ship, twelve 12½-pounder, three 3-pounder and six Maxim guns. Finally, the *Katori* has five submerged tubes for the firing of 18-inch torpedoes.

The armor on the new ship is arranged on the system first introduced into the *Mikasa*, ten of the 6-inch guns being in a concentrated casemate amidships instead of in a series of separate casemates. The whole of the broadside, from some distance below the water line to the upper deck, is armored. The main belt is 9 inches, reduced gradually to 4 inches towards the stem, and to 3½ inches at the stern. The minimum thickness of the upper parts of the broadside armor is 6 inches. Armored bulkheads are carried across the ship at the forward and aft ends of the citadel. Ten of the 6-inch guns are placed within this 6-inch armor, but each is isolated from its neighbors by transverse and longitudinal armor bulkheads. On the upper deck there has been constructed for the two remaining 6-inch guns a special armored battery. These guns have a very advantageous position for dealing with torpedo attack. The 12-inch gun barbettes have armor 10 inches in thickness generally, and those for the 10-inch guns have 6-inch armor. The ammunition hoists of all the guns are armored.

The third element in design has reference to the speed, and in this case the specified rate is 18½ knots; but as the *Mikasa* very considerably exceeded her designed rate, it would not be surprising if in this case also 19 knots is realized. The propelling machinery is of the type now usually adopted in battleships, there being two sets of four-cylinder triple-expansion engines, balanced on the Yarrow-Schlick-Tweedy system. The high-pressure cylinder is 35½ inches, the intermediate 56 inches, and the low-pressure cylinders each 63 inches in diameter, the stroke in all cases being 48 inches. It is anticipated that with 120 revolutions per minute and steam at 200 pounds pressure, the engines will develop 16,000 indicated horsepower. The steam pressure at the boilers is to be 230 pounds per square inch. The engines are designed to turn the pro-

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pellers inwards when going ahead, and the starting platform is in the center of the ship. Wrought-steel columns form the supports of the cylinders, which are independent castings, while the back supports are of the ordinary cast-iron A-frame, with ample slipper-guide surfaces. This gives a fine, open engine. Stephenson link motion is adopted for working the valves, which are of the piston type on the high-pressure and intermediate cylinders, and of the double-ported flat design on the low-pressure cylinders. The crank shaft is in two interchangeable pieces, and the propeller shaft is 18 inches in diameter, with a 10-inch hole. The propellers have four blades, the diameter being 17 feet 3 inches; they are made entirely of bronze. There are four condensers, placed in the wings of the ship, the collective cooling surface being 17,000 square feet. As to the boilers, these are of the latest improved Niclausse type, twenty in number, located in three separate boiler rooms. The heating surface is 44,000 square feet—a ratio of 2.75 square feet per indicated horsepower. The total grate area is 1,334 square feet. There is, of course, a complete system of pumps, with the addition of independent air pumps, which are now almost universally adopted in connection with large high-power marine installations.

The total coal capacity is 2,100 tons, which will give the ship a high radius of action. Special attention has been given to the ventilation and general comfort of the living quarters of the ship. The thermo-tank system of ventilation has been fitted for the cabins and crew spaces, and for ventilating and cooling the magazines and shell rooms. A complete system of steam heating is adopted for the officers' quarters and crew spaces, and electricity is used in connection with ventilation as well as for lighting the ship. The vessel will have a total complement of 980 officers and men.

—"Engineering," London.

PERU.

The Peruvian Government has placed an order with Messrs. Vickers, Sons & Maxim for a cruiser of 24 knots speed closely

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resembling, but slightly larger than, the scouts *Sentinel* and *Skirmisher*, which were built by the company for the British Navy.

RUSSIA.

New Ships.—The Russian battleship *Andrei Pervozvannyi*, of 16,630 tons, was launched at the Galerny Island yard, St. Petersburg, recently, and a sister ship, the *Pavel I*, is in hand in the Baltic yard. A great deal of uncertainty surrounds the new Russian ships, but a number are building from funds raised by private subscriptions. One of these, the torpedo cruiser *Dobrovoletz*, was launched at the Putiloff yard on June 11. The "Yacht" says that Russia is preparing to build, or actually building, three armored cruisers of the *Bayan* class—the *Admiral Makaroff*, *Pallada* and a new *Bayan;* as also four gunboats of the *Gyliak* class, named after the vessels lost—*Gyliak*, *Korietz*, *Sivoutch* and *Bobr*. Ten river gunboats are also mentioned—the *Oratschanin*, *Mongol*, *Buriat*, *Saboikaletz*, *Sibiriek*, *Koreia*, *Kirgis*, *Kalmuk*, *Amuretz* and *Ussurietz;* two destroyers of 300 tons, the *Lietnant Melejeff* and *Inzhener Mechanik Anastasoff*. In addition to these the names of twenty other destroyers are given, some of them being those of destroyers lost in the present war, and others named after officers who have distinguished themselves—Captain Jurassovky, Lieutenant Sergeieff, and Engineers Sverieff and Dmitrieff.—"Army and Navy Journal."

SWEDEN.

Magne—*Launch of.*—The torpedo-boat destroyer *Magne*, for the Royal Swedish Government, was launched on Wednesday, August 2d, from Messrs. Thornycroft's Church Wharf Works. She is a vessel of the *Shirakumo* type, which has proved so successful in the present Russo-Japanese War. Her guaranteed speed is 30½ knots, to be maintained for three hours with a load of 55 tons on board. The vessel is 219 feet 9 inches long over all, 20 feet 9 inches beam, and has a draught of 8 feet 3 inches at the screws. The machinery, which is over 7,200 indicated horsepower, consists of two sets of four-cylin-

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der vertical triple-compound condensing engines and four Thornycroft water-tube boilers, worked under forced draft on the closed-stokehold principle. The bunkers have a capacity for 80 tons of coal, giving a radius of action of 2,800 nautical miles at a speed of 15 knots per hour. The armament consists of six 57-millimeter quick-firing guns—one mounted on the conning tower, one aft, and four on the broadside. There are also two 45-centimeter revolving torpedo-tubes. The boat is extensively sub-divided into watertight compartments by transverse bulkheads and watertight decks under the living spaces.—"Engineering," London.

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MERCHANT SHIPS.

Viking.—The new turbine steamer *Viking*, built for the Isle of Man Steam Packet Company by Sir W. G. Armstrong, Whitworth & Co., completed her trial the latter part of June.

The principal dimensions are: Length between perpendiculars, 350 feet; length over all, 361 feet; breadth, extreme, 42 feet; depth to main deck, 17 feet 3 inches; depth to shelter deck, 25 feet 3 inches.

The subdivision of the vessel into watertight compartments has received special attention, and the principle has been carried so far that the ship will float with any two compartments full of water. The vessel has been built to Lloyd's requirements, and to conform with all the rules of the Board of Trade for a passenger certificate. To meet the special requirements of this trade she is designed to carry a very large number of passengers—nearly 2,000—and to afford them ample accommodation in the way of promenade space, together with extensive facilities for refreshment in the shape of numerous dining saloons and bars. Her shelter deck extends uninterruptedly from stem to stern, a length of 361 feet, and above it is a big promenade deck, about two-thirds this length, which is carried out to the ship's sides, and is thus 42 feet wide at its broadest point. The first-class accommodation is aft and the second-class forward, the middle part of the ship being mainly occupied by the boilers and engine room. The main, upper and shelter decks are continuous, except in way of the engine and boiler rooms.

Abaft the main entrance hall, on the shelter deck, there are four private cabins and a tea room, close to which there are two of the companion ladders leading to the promenade deck above.

The main saloon is the full width of the vessel, and is

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lighted by skylight and large rectangular windows at the sides. The woodwork is in dark oak, and a series of black-and-white reproductions of famous pictures, by well-known artists, forms a pleasing contrast with the dark tones of the somewhat severe paneling, which is also broken by the deep-set window openings, and relieved by a simple carved design. The entrance hall is paneled in oak, of the same tone as the main saloon, combined with blue and white Dutch tiles.

The ladies' room is entered through folding doors on the port side of the main saloon. It is paneled in sycamore in simple style, and is lighted by large windows in the ship's side and by the large light and air trunk which leads to the dining saloon on the deck below. A large lavatory opens off the ladies' room.

At the after end of the main saloon a stairway leads down into another saloon, where accommodation is provided for those who wish to lie down.

The dining saloon is entered by means of a stairway alongside the main entrance to the saloon. It provides seating accommodation for about 100 people, and is decorated with oak paneling. The usual pantry arrangements, with hot plates, lift to the galley, &c., are provided, and all the accessories required to make the service in this department as expeditious as possible.

The first-class smoking room is on the shelter deck, and is entered by doors placed just forward of the stairway from the shelter to the upper deck.

The second-class accommodation is at the forward end of the vessel, and consists of two large general saloons, one of which is arranged for dining, and a ladies' saloon. Lavatory accommodation is provided in convenient places throughout the vessel.

The propelling machinery consists of three sets of turbines manufactured by the Parsons Marine Steam Turbine Company. The high-pressure turbine is in the center, with a low-pressure turbine on each side. The surface condensers are in the wings at the after end of the engine room, with the centrifugal

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circulating pumps close alongside them. Two large independent air pumps, of Messrs. Weir's make, are fitted conveniently near the condenser, and discharge into a large feed tank, whence the feed water is led to the automatic float-control tank, and so to the two feed pumps, which are also of Messrs. Weir's make. These pumps discharge to the boilers direct, or through an exhaust feed heater on Parsons' patent, which gave very satisfactory results on the trials. The Parsons' patent vacuum augmenter is fitted in connection with the main condenser, enabling a high vacuum to be maintained. The lubrication of the bearings is effected by means of an oil-circulating pump, which drives the oil, under considerable pressure, through all the bearings, and into a cooling and filtering apparatus on its way back to the pump. The stern-way turbines are arranged as usual within the low-pressure casings, and revolve idly *in vacuo* while the vessel is steaming ahead. The three shafts are of Armstrong-Whitworth Openshaw steel, and the propellers of a special bronze mixture worked up to a highly-polished surface.

Steam of 160 pounds pressure is supplied by four double-ended boilers, 15 feet in diameter and 19 feet 6 inches long, placed in two boiler rooms separated by a watertight bulkhead. The stokeholds are arranged for forced draft, and the air is supplied by four fans placed in a separate compartment on the shelter deck, and communicating with the stokeholds by means of straight air trunks. An emergency exit from each boiler room is arranged through the air lock into the fan room. The funnels are fitted with outer casings which are utilized as upcast ventilators, and for this purpose are connected to compartments in various parts of the ship. A spark arrester, Messrs. Denny & Brace's patent, is fitted in each funnel. The result of its trial was most satisfactory, as there was no sign of sparks or cinders when running at the maximum speed. The whole of the stokehold and boiler-feeding arrangements were carried out by the Wallsend Slipway and Engineering Company.

Two sets of trials were carried out, irrespective of the

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builder's preliminary test. The first official trial took place over the measured mile off the mouth of the Tyne, under somewhat unfavorable weather conditions. There was a fresh northeasterly breeze and from 2 feet to 3 feet of sea. The air pressure in the stokeholds did not exceed $\frac{3}{4}$-inch water column, and no attempt was made to develop the maximum power of the boilers. On June 14 the vessel was taken to sea for an extended trial. A course was set from Souter Point to Flamborough Head, 68.277 nautical miles, and the distance was traversed in both directions, care being taken to eliminate, as far as possible, the effect of the tide. The southward run was made at a speed of 23.684 knots, and the northward at 23.375 knots, or a mean speed over the whole distance of 23.53 knots. The vessel was kept at full speed for nearly seven hours. Although the power developed on this trial was higher than on the preliminary trials, there was very considerable reserve in the boilers.

The maneuvering qualities of the ship are very favorable.

Wm. E. Corey—*Launch of.*—The steamer *Wm. E. Corey*, building for the Pittsburg Steamship Co., was launched from the South Chicago yard of the American Shipbuilding Co. on June 24th.

The *Corey* is the second of the quartet of giant steamers building for the Pittsburg Steamship Co. to be launched. She differs from the others only in cabin arrangement. The other three ships have no passenger accommodations whatever, having only one spare room to accommodate an occasional representative of the company. The passenger accommodations of the *Corey*, however, are quite complete, the staterooms, five in number, being decorated in birch and enameled white, each stateroom having a private bath. There is also an observation room on the forecastle deck with unusually large windows. This room is decorated in Circassian walnut. The passenger accommodations, of course, include a private dining room. The *Corey* is, of course, the flagship of the fleet.

The *Corey* and her sisters offer further testimony to the originality of design now marking the evolution of shipbuild-

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ing on the lakes. They are in a number of features unlike any ships built or building. In the first place, they are the longest on the lakes, being nine feet longer than the *Augustus B. Wolvin*, or 569 feet over all, 549 feet keel, 56 feet beam and 31 feet deep. While the general principle of construction is the same as their predecessors, in the absence of stanchions and stringers or any interference below deck with the unloading machines, the vessels are not of arch construction. Mr. Colby, after consulting a number of authorities, concluded that an arch was primarily intended to support weight from the top, and that a more rigid longitudinal construction could be secured by a straight girder across the deck.

The construction of the cargo hold also is a departure from anything at present on the lakes. The cargo hold descends in a straight line from the deck until within 9 feet of the tank top, when an inner hold or hopper is constructed 9 feet from the skin of the ship and descends to the tank top. The top of the hopper is joined to the skin of the ship by an incline construction, thus avoiding what would otherwise constitute a shelf and also preventing any lodgment of ore. The water bottom is 5 feet 6 inches deep, and is used for water ballast in conjunction with a 9-foot space formed by the hopper and the sides of the ship. This construction gives enormous water-ballast capacity, approximating 8,500 tons, making the steamer navigable in all sorts of weather and providing extraordinary rigidity in the longitudinal section. The hatches number thirty-four, and are spaced 12-feet centers, the hatch covers being operated by the Brosseau device, enabling the hatches to be opened or closed from a central station. These ships are designed for a carrying capacity of 10,000 tons, but when favored with deep draught their actual cargoes are likely to be considerably in excess of this figure, as has already been proved by the initial trip of the *Gary*.

The engines are triple-expansion, with cylinders 24, 39 and 65 inches diameters by 42 inches stroke, supplied with steam from two Scotch boilers, 15 feet 4½ inches diameter by 11 feet 6 inches long, fitted with Ellis & Eaves induced draft and

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allowed 170 pounds pressure. It is interesting to note that this is precisely the engine equipment of the *Manola*, which was built by Pickands, Mather & Co. in 1890, and which was at the time the largest carrier on the lakes. The *Manola* would carry in a full season 68,000 tons of ore. The *Corey* will carry on an equivalent number of trips over 225,000 tons of ore with the same fuel consumption. It is clear from these figures that the economical carrier is the single steamer of large carrying capacity and low power. The day of the consort is past.—Extract from "Marine Review."

The Fore River Shipbuilding Company, Quincy, Mass., closed a contract with the Southern Pacific Railway for a freight and passenger steamer for the Morgan line, to be fitted with twin screws and Curtis turbines. The boilers will be supplied by the Babcock & Wilcox Company. The principal dimensions of the new steamer will be as follows: Length over all, 440 feet 6 inches; length between perpendiculars, 415 feet 8 inches; beam, molded, 53 feet; depth molded to upper deck at side, 37 feet; depth molded to main deck at side, 29 feet; deck molded to lower deck at side, 20 feet 3 inches; height between saloon deck and upper deck, 7 feet 9 inches; height between promenade deck and saloon, 7 feet 6 inches; height between saloon deck and boat deck, 7 feet 6 inches; length of the amidship superstructure, about 182 feet; displacement, loaded, about 10,000 tons.—"Marine Review."

Turbine Steamers.—The W. & A. Fletcher Company, of Hoboken, N. J., has recently secured the contract for three turbine steamers. One of these, with 5,000 horsepower motors, is for the Eastern Steamship Company, and is intended for service between Boston, Mass., and Halifax, N. S. The others, of 10,000 horsepower each, are for the Metropolitan Steamship Company; they will run from New York to Boston.

Onward.—This turbine steamer, built by Messrs. Wm. Denny & Bros., Dumbarton, to the order of the Southeastern and Chatham Railway Co., proceeded to sea on her official trial trip. On the mile a mean speed of 22.54 knots was ob-

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tained, a record speed for passenger turbine steamers, whilst with the turbines reversed there was a backward speed of 15½ knots. This latter feature, combined with the increased ahead speed, will mean a considerable saving of time in the channel passage. The official trial trip of the vessel consisted of a continuous run of six hours, when everything worked satisfactorily.

Invicta.—This turbine steamer, built by Messrs. Denny Bros., Dumbarton, to the order of the Southeastern and Chatham Railway Co., proceeded to sea on her official trial trip recently. On the measured mile at Skelmorlie her mean speed for a double run was 22.935 knots, or practically 23 knots. The contract condition was that one of the boilers should be out of action—that is to say, five-sixths of the boiler power was to be available—and under these conditions the speed attained was 21.85 knots. During this trial, too, the electric machinery, which is driven by a small steam turbine, was working at test pressure. As the guarantee speed for the four-hours' run was 21 knots, this trial also proved most satisfactory to the builders. The *Invicta's* turbines and auxiliary machinery, as well as the boilers, were constructed and fitted on board by Messrs. Denny & Co.

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OBITUARY.

CAPTAIN THEODORE F. BURGDORFF, U. S. NAVY (RETIRED).

It is with regret that we announce the death of Captain Theodore F. Burgdorff, U. S. Navy (Retired), a member of this Society, at the U. S. Naval Hospital, Philadelphia, Pa., on July 29, 1905.

Captain Burgdorff was born on November 22, 1854, at Newark, N. J. He was appointed a Cadet Engineer ta the U. S. Naval Academy in 1873, was graduated in 1875, and on July 1, 1877, was commissioned an Assistant Engineer in the Navy.

On June 22, 1886, he was promoted to Passed Assistant Engineer, and to Chief Engineer on February 26, 1897. Upon the passage of the Naval Personnel Bill, March 3, 1899, his rank was changed to that of Lieutenant Commander, and on September 30, 1904, he was promoted to Commander.

Shortly before his death Captain Burgdorff made application for voluntary retirement, and on June 30th was placed on the retired list of the Navy with the rank of Captain.

His first cruise was made on the *Swatara*, at the expiration of which he was detailed for duty in the Bureau of Steam Engineering; after a cruise on the *Nipsic*, he was again ordered to the same Bureau.

Captain Burgdorff afterwards served on board the *Iroquois*, *Thetis*, *Oregon* and *Monadnock;* in 1903 and 1904 he served as Fleet Engineer of the South Atlantic Station on board the flagship *Newark*.

He was at various times detailed for duty on shore at the New York and Mare Island Navy Yards, and was for several years the Instructor of Engineering at the University of Tennessee.

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Upon the completion of his cruise on the *Newark*, in October, 1904, he was ordered to inspection duty under the Bureau of Steam Engineering, and remained upon this duty up to the time of his death.

Captain Burgdorff ranked high in his profession, and in his death the Society and the Naval service have suffered a distinct loss.

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BOOKS RECEIVED.

THE NAVAL POCKET BOOK FOR 1905.—By SIR W. LAIRD CLOWES, London; W. THOCHER & Co., 2 Creed Lane, E. C. Price, 7*s.* 6*d.* net.

This little book contains, in convenient form, much valuable information concerning the navies of the world.

Having been before the public for ten years, there is little need for further recommendation.

The following is a summary of the contents, the information given being brought up to date: Calendar and Naval Remembrances, 1905, Classification of Ships, Comparative Summary of the Fighting Fleets, Classified List of the Navies of the World, Guns and Small Arms, Notes on Torpedoes, Naval Personnel and Comparative Rank, Retiring Ages in European Navies, List and Particulars of Dry Docks, Miscellaneous Tables, Trial-Trip Tables, Plans of Ships.

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JOURNAL
OF THE
AMERICAN SOCIETY OF NAVAL ENGINEERS

ERRATA.

No. 4, Vol. XVII.

Page 1104.—Eighth line from top, change "Pressures 0 to 1,000 pounds," to read "Pressures 0 to 100 pounds."

Same page, fifteenth line from top, change "boat circle" to read "belt circle."

Page 1110.—Second line from bottom, change "fan" to read "flap."

Page 1133.—Thirteenth and fourteenth lines from the top, omit the words "complement of the."

Cut out, and insert in No. 4, Vol. XVII.

In proof, there can be cited the name of Tournaire, Inspector General of Mines in France, whose theoretical description of a steam-turbine with multiple reaction wheels, addressed in 1853 to l'Académie des Sciences, is startling when compared with the engine constructed by Parsons in England 34 years

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JOURNAL

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The Society as a body is not responsible for statements made by individual members.

COUNCIL OF THE SOCIETY
(Under whose supervision this number is published).
Rear Admiral GEORGE W. BAIRD, U. S. N., Retired. Commander J. K. BARTON, U. S. N.
Captain A. F. DIXON, U. S. N. Commander W. M. PARKS, U. S. N.
Lieutenant CHAS. K. MALLORY, U. S. N., Retired.

THE RATEAU STEAM-TURBINE AND ITS APPLICATIONS.

BY M. J. REY.

Translated from the April, 1904, issue of the Memoirs and Compte Rendu of the Works of the Society of Civil Engineers of France, by Chief Engineer B. F. ISHERWOOD, United States Navy, Honorary Member.

The substitution of the steam-turbine for the reciprocating piston steam-engine, which has been realized in the present epoch, is the most important fact in the history of steam-motors, if not the greatest step in the progress of the mechanical arts, since the memorable invention of Watts.

Numerous precursors, sagacious inventors, penetrating intellects, of the early years of the XIX century, foresaw the future reserved for the rotative movement and continuous flow as opposed to the intermittent flow and alternate movement alone employed up to almost the present time.

In proof, there can be cited the name of Tournaire, Inspector General of Mines in France, whose theoretical description of a steam-turbine with multiple reaction wheels, addressed in 1853 to l'Académie des Sciences, is startling when compared with the engine constructed by Parsons in England 34 years

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afterwards. Also the name of Pilbrow, who, in a patent taken out by him in 1843, described a steam-wheel furnished with convergent-divergent distributors on the principle embodied later in the well known steam-turbine of Laval in Sweden.

To realize and to utilize the steam-turbine, certain conditions are indispensable:

1st. The exact knowledge of the physical properties of steam, as well as of the thermo-dynamic laws enabling the calculation to be made of the velocity of the flow of steam, and of the total work it can do for a given fall of pressure.

2d. The possibility of using metals of great strength and perfect homogeneity, capable of sustaining the enormous strains caused by centrifugal force in revolving bodies at the high angular velocities produced.

3d. A new mode of transmitting the power of the turbine to the varied receiving machines used in industrial operations that will utilize, without notable loss, its great velocity.

The progress of the physical sciences, especially the progress of metallurgy, and the discovery of the electrical generatrices and motors of induction, have now given a practical solution of this great problem, and have rendered possible this revolution in industrial mechanics.

At the passing moment, to say that the steam-turbine will be the steam-engine of the future, would not be correct: it is already the steam-engine of the present. Several hundreds of thousands of horsepowers of it are now either in use or in construction; and there is no temerity in the statement that with the exception of locomotives, lifting engines, rolling engines, and, in a certain sense, marine engines, there will be no more building of reciprocating steam-engines.

There is no intention to here give a resumé, however succinct, of the efforts which have been made by engineers and specialists to produce the steam-turbine of these latter years. To treat the subject as a whole, the numerous patents of the precursors* would have to be passed in review and examined;

*See, notably, the interesting investigation: *Roues et Turbines à Vapeur*, published by Mr. Sosnowsky in le Bulletin de la Society d'Encouragement à' l'Industrie Nationale. 5th Series, tome II, page 1153 and following.

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certainly, several hundred of them at the present time taken out in the different principal countries during the last fifteen years.

The intention is much more modest; it will be restricted to a monograph of the steam-turbine created by Mr. Rateau, Engineer in the Corps of Mines, in collaboration with the firm of Saütter, Harlé & Co., which has made from the beginning all the investigations and all the experiments concerning the new type of engine.

In order to render the question precise, a rational classification is indispensable of the different systems of steam-turbines, so as to be able to state the exact category to which the Rateau turbine belongs, its principle of construction, its mode of functioning and its technical advantages.

Previous to making this classification, a relation of some historical facts regarding the invention itself, would seem to be useful.

HISTORICAL FACTS RELATING TO THE RATEAU TURBINE.

The principal works of Mr. Rateau on steam-turbines, go back to the year 1894, in which he commenced the construction of a steam-turbine with one rotating wheel for driving dynamos.

But, before this epoch, Mr. Rateau had been occupied with the question of steam-turbines, and there will be found in the monthly comptes rendus of the Societé de l'Industrie Minerale (seances of March 1st and 12th April, 1890), an interesting communication from that engineer concerning the Parsons turbine, as well as a discussion of the use of steam in mass motion or under pressure.

The fact is important that Mr. Rateau described at that epoch, the construction of steam-turbines with wheels and distributors moving in opposite directions, an idea which has recently been taken up again by different inventors.

The investigations of the steam-turbine of one wheel were pursued during the years 1895 and 1896, at Saint-Etienne and at Paris. After having tried successively the centrifugal method and the centripetal method, they resulted in the adop-

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tion of groups of electrogens of one or two dynamos with helicoidal wheels of the Pelton system, of which a small number were constructed and are still in use.

Mr. Rateau very soon perceived the disadvantages of the one wheel turbine, and the advantages of a system of multiple wheels, the working steam acting by impulsion instead of by the method of reaction employed by Parsons. His ideas took definite form towards the year 1897, and during that epoch he conceived the principle of his helicoidal multicellular turbine of impulsion, the merits of which will be shown later on. The following year the firm of Sautter-Harlé commenced the construction of a steam turbine of 900 horsepowers based on this principle.

Of the numerous difficulties encountered in the course of the construction, and developed by the first experiments, only a few need be recalled: they were those of obtaining the perfect equilibrium required in the rotating pieces; those of effecting the automatic lubrication of the pillow-blocks; those of the packing of the stuffing-boxes of the shaft; those of the regulation of the velocity; those of the best arrangement for reducing the friction of the discs upon the steam; and those of other details that would require too much space for description; demanding a great deal of thought and time to overcome them and perfect the practical working of the engine.

A first unit of 1,000 horsepowers was intended for the Exposition of 1900. It was not ready in time, and the first experiments were made with it after the closing of the Exposition.

While the turbine was being created, the generator of electricity that it was to operate had also to be created. The triphase current was attacked, and there was constructed a first alternator of iron revolving with a power of 700 kilowatts.

From the year 1901 the progress was rapid.

The employment of low pressure turbines also increased the field of action of these new machines. At the present time the firm of Sautter-Harlé have either delivered or have in construction more than 17,000 horsepowers of turbines, and this figure is rapidly increasing.

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Before terminating what was to be here said about the theoretical work of Mr. Rateau, there must be added that he has deposited in the Academy, a dossier containing all his researches up to that period on steam-turbines.*

CLASSIFICATION OF DIFFERENT SYSTEMS OF STEAM-TURBINES.

Steam-turbines constitute a division of the general class of engines in which a fluid does external work by acting upon one or more rotating wheels by means of buckets, chambers, or passages made in or upon them. [Bucketing.]

In the steam-turbine the rotating wheel is pushed by the motor-fluid, the appropriate direction of which is given by distributing passages or canals.

The steam-turbine, or an element of it, is consequently composed of two fundamental organs: the rotating wheel and the fixed distributor.

The rotating wheel can be formed in a number of ways. According to the case, it may be one massive wheel in a single piece dressed from a block of metal and having its buckets cut in it. Or it may be of plate iron with its buckets affixed to it. Or it may be a solid wheel on the periphery of which its buckets made in two separate bands may be shrunk on.

The inventors have free license, and many methods are possible; but, in reality, there are only two prevailing types, namely, the light wheel of plate iron with buckets applied,

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* The different notes which this dossier contained, were those already published, and others not yet edited. They relate to all the interesting subjects concerned in steam-turbines. Their names are as follows:

1st. *Report on Steam-Turbines* made to the International Congress of Applied Mechanics, in 1900.

2d. *Treatise on Turbo-machines* (published in 1900).

3d. *Abacus of the theoretical consumptions of a steam-engine* (published in the Annales des Mines of 1897).

4th. *Resistance of Revolving Bodies* (published in the Bulletin de la Societé de l'Industrie Minerale of 1890).

5th. *Apparatus for measuring the humidity of steam* (Annales des Mines, 1898).

6th. *Memoir of Experimental Researches concerning the Efflux of Steam* (inedited, since published in the Annales des Mines, 1902).

7th. *Theory of Steam-Turbines* (inedited, since published in the Revue de Mechanique, 1903).

8th. *Experiments on the impulsion of steam* (inedited).

9th. *Experiments on Steam-Turbines* (inedited).

10th. *Experiments on a high pressure centrifugal ventilator* (inedited, since published by the Societé d'Encouragement, and in the Bulletin de l'Industrie Minérale, November, 1901).

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and the solid wheel with its buckets cut out of the same block.

A third method of construction can be adopted, consisting in the suppression of the body, properly so called, of each wheel, and mounting the different bands of buckets upon a single drum.

The distributor is formed, as in the case of hydraulic turbines, by curved directrices; or it is composed simply of one or of several tuyeres of circular cross section, calculated according to the properties of the expansion of elastic fluids.

The construction of distributors can be very much varied; sometimes canals or passages are cut into the mass of the metal; or are formed, on the contrary, by separate small pieces of metal applied to it; and, according as the distribution is made over all the periphery, or over a fraction of it, the construction can be varied in many ways.

The classification of steam-turbines can be made according to three kinds of consideration:

1st. According to the number of rotating wheels composing the turbine.

2d. According to the route followed by the motor-fluid.

3d. According to the mode of functioning of the motor-fluid.

These three categories will now be successively passed in review, and examined in order to ascertain the different kinds of turbine represented by each in its turn.

NUMBER OF ROTATING WHEELS.

The classification of the number of rotating wheels, comprises two kinds:

The turbine with a single rotating wheel;

The turbine with multiple rotating wheels.

The turbine with a single rotating wheel is divided, in its turn, into a turbine with the buckets applied, and a turbine with the buckets cut out of the same block of metal as the wheel.

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Among the single wheel turbines with applied buckets may be particularly cited the Laval turbine and the Seger turbine.

The Rateau turbine of one wheel, constructed in 1894, has the buckets cut out of the mass of the metal, the whole wheel being in a single piece. It is the same with the Riedler-Stumpf turbine of which so much has been said in Germany during the last two years.

The multiple wheel turbine comprises the following three kinds:

The drum turbine;

The turbine with groups of wheels;

The multicellular turbine.

Among the drum turbines, the best known and the most extensively used is the Parsons turbine.

The turbines with groups of wheels—also termed turbines with speed in stages—appear for the first time with the Curtis turbine.

As for the multicellular turbines, the best known, actually in use, is the Rateau turbine.

ROUTE FOLLOWED BY THE MOTOR-FLUID.

The route followed by the motor-fluid in the rotating wheel can be determined in two principal ways.

In the first case, the motor-fluid traverses the wheel parallel to its axis. The axial or helicoidal turbines are thus obtained.

During this route the fluid continues sensibly at the same distance from the axis of the wheel.

In the second case, the motor-fluid moves in a plane perpendicular to the axis of the wheel. The radial turbines are thus obtained, and they comprise two kinds. The centrifugal, in which the fluid moves from the axis. The centripetal, in which the fluid moves towards the axis. Finally, these two modes—the axial and the radial—can be combined by causing the fluid to move obliquely in relation to the axis, and in this way will be obtained the helico-centrifugal or helico-centripetal turbine.

As examples of the helicoidal turbine, and of the axial tur-

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bine, there may be cited all the steam-turbines employed in practice.

The mode of centrifugal route, or of centripetal route, has given, notwithstanding numerous trials, but very mediocre results.

The radial turbines are a great deal more complicated in construction, and they do not give any advantage either in fuel economy or in mechanical functionment over the helicoidal turbines.

MODE OF FUNCTIONING OF THE MOTOR-FLUID.

The third category for the classification of steam-turbines is based on the mode of the functioning of the motor-fluid.

As, in the case of hydraulic turbines, all steam turbines can be made to function either by reaction or by impulsion.

The first group, or reaction turbines, comprehends notably the Parsons turbine and the Westinghouse turbine, which is only a variant of it.

In the second group are found the turbine of Laval, the turbine of Rateau, the turbine of Curtis, the turbine of Riedler, and the turbines of many others.

I.—CLASSIFICATION BY THE NUMBER OF WHEELS.

There will now be passed in review, a certain number of examples chosen from the best known steam-turbines, in order to illustrate the classification made, and to enable it to be better comprehended.

TURBINE OF ONE WHEEL.

Figure 1 represents the well known helicoidal wheel of Laval's steam-turbine.

There will be remarked the distributors formed of converging-diverging tuyeres placed obliquely to the wheel, which is formed of one solid piece with applied buckets shaped like the tail of a swallow, and held by a metallic felly.

By using a special steel employed by the firm Bréguet in the construction of this wheel, its peripheral velocity can, in practice, be pushed up to 1,150 feet per second; in some experiments 1,300 feet were obtained.

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There may here be recalled that the Laval wheel is mounted upon a shaft of small diameter whose flexibility allows the axis of inertia of the wheel to follow the geometrical axis of the wheel notwithstanding the small defects of equilibrium impossible to be suppressed in practice.

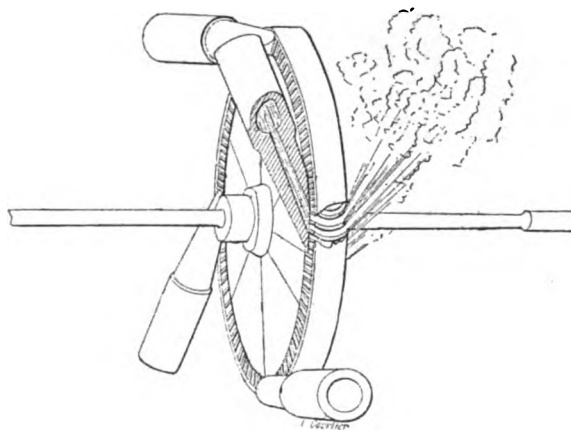

Fig. 1.—LAVAL TURBINE.

Figure 1, Plate I, shows the helicoidal wheel of a Rateau turbine of one wheel on the Pelton system.

This system, patented in France in 1894, is constituted essentially of a wheel with its buckets cut out of a single block, instead of the buckets being made separately and attached. This construction assures greater strength to the rotating wheel. The wear by the friction of the buckets does not cause any more inconvenience, because their thickness is much greater. There is no more danger of the wheel being thus thrown out of equilibrium by the suppression or the diminution of one or of several buckets.

With a wheel of this construction of 1 foot diameter, it is possible to make 24,000 revolutions per minute which represents a peripheral velocity of, say, 75,400 feet per minute.

As has been remarked, the buckets have the form of those on the Pelton wheel; consequently, the distribution of the steam is made diametrically, and is perfectly symmetrical as

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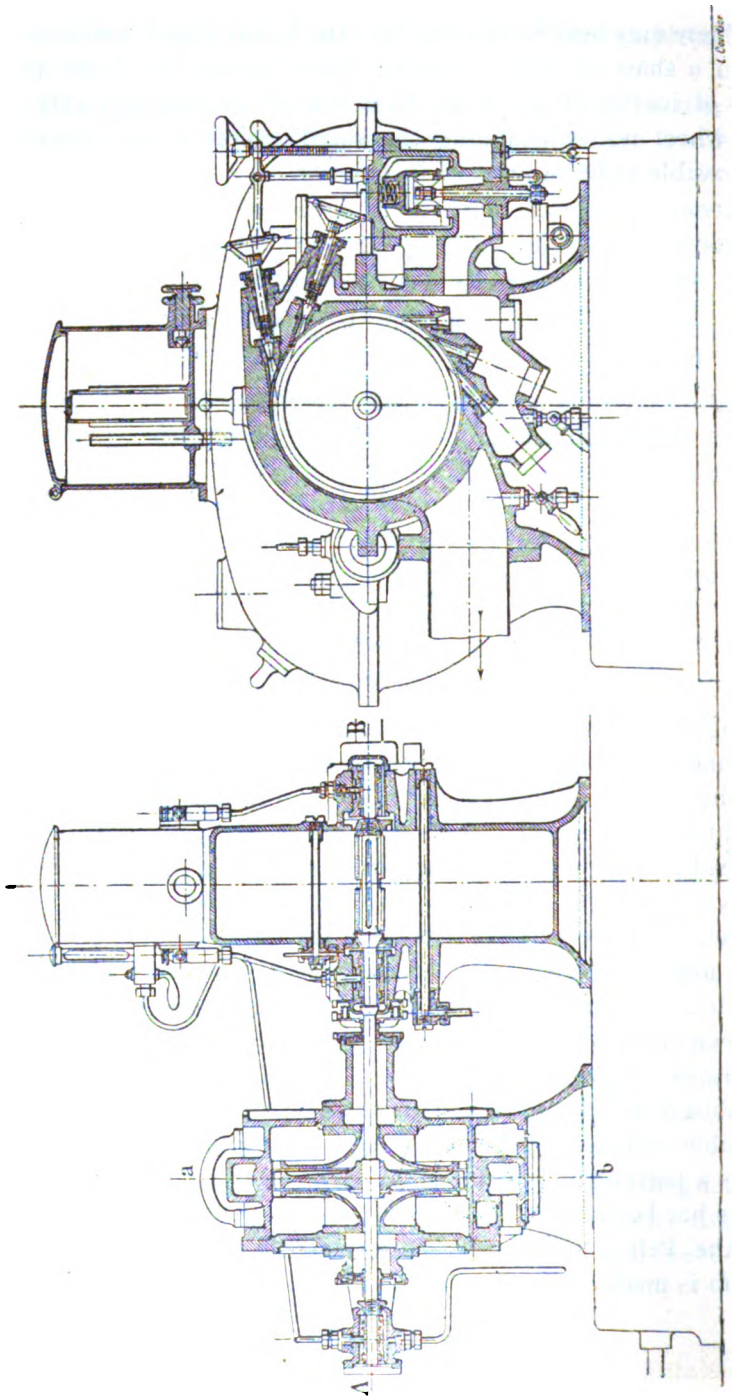

Fig. 2.—Turbine of One Wheel, Rateau System.

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regards the wheel, which is thus free from any longitudinal or axial push.

The combination of the wheel and the distributors, which latter are formed of diametrically placed tuyeres, is shown in Figure 2, which represents an electrogen group of 40 kilowatts power at 15,000 revolutions per minute, constructed by the firm of Sautter-Harlé and Company in 1895.

Several examples of these turbines are in operation at the present time.

Figure 2, Plate I, represents one of them. The distribution by diametrically opposite tuyeres is plainly visible. The shaft of the turbine carries a pinion operating by two sets of helicoidal gearing, the two shafts of the dynamos.

This arrangement is the same as that which is employed in the Laval turbine. It has also the same disadvantages in the transmission of the power from the turbine to the dynamos.

During these last years the construction of the wheel with buckets according to the Pelton system and designed by Mr. Rateau, has been again put in practice in Germany, by Professors Riedler and Stumpf for a one wheel turbine constructed by the Allzemeine Elektricitats Gesellschaft.

Figure 3, Plate I, represents a Riedler and Stumpf wheel for a turbine of 2,000 horsepowers. This figure is extracted, as well as the following ones, from a Memoir by Riedler to the German Technical Society for the construction of vessels.

The distributors employed for the Riedler and Stumpf turbine, are represented in section and in external view in Figure 4, Plate I.

Attention is called to the great number of tuyeres required to form a distributor for the Riedler and Stumpf wheel. For great powers, there must be delivered on the same wheel a correspondingly large weight of steam, and it is impossible to apply to the periphery of the wheel larger tuyeres and fewer in number, because their length would become too great.

Inventors soon comprehended that a turbine with but one wheel, could have only a mediocre development of power, and they have adopted for it two wheels the general action of

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which is represented for a turbine of 2,000 horsepowers in Figure 3.

There will be described later, the different causes that affect the mechanical rendering in a steam-turbine, but there can be easily shown immediately the advantage of multiplying the number of wheels to fraction the fall of the steam-pressure.

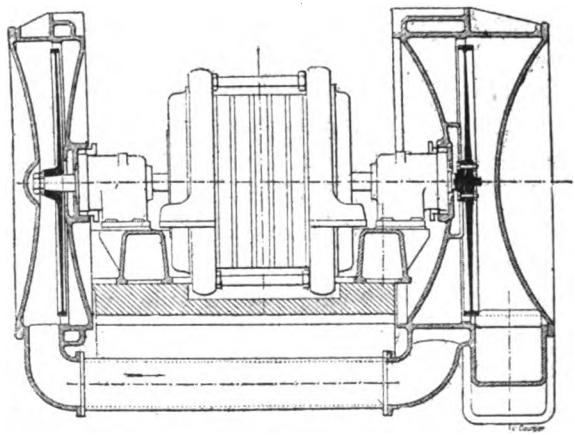

Fig. 3.—TURBINE OF TWO WHEELS, RIEDLER.

The mechanical rendering of a steam-turbine depends essentially on the ratio of the peripheral speed of its rotating wheel to the absolute speed of the steam at the moment of introduction into the turbine. With the considerable speeds of elastic fluids that result from a fall in their pressure, there would be required either enormous peripheral speed for the rotating wheel of the turbine or a fractionment of the total fall of pressure into a series of smaller falls.

If steam of the effective pressure of 142.25 pounds per square inch be used, expanding into a vacuum of 25.58 inches of mercury, the speed of the efflux for saturated steam will be 3,724 feet.

To obtain an acceptable mechanical rendering with a one wheel turbine, the peripheral speed of the wheel must exceed 1,312 feet, a limit very difficult to attain for wheels of large diameter. Thus, for a turbine of 1,000 kilowatts, for example,

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making 1,500 revolutions per minute, the diameter would have to be at least 16.4 feet for the peripheral speed required. With such dimensions, the construction of a wheel with the exactness needed for the perfect equilibrium of all its parts, would be nearly impossible. Therefore, it is not astonishing that the inventors of single wheel turbines have successively increased the number of their elements.

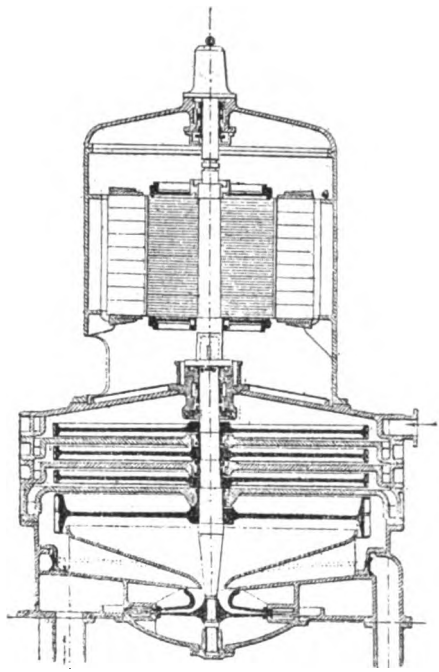

Fig. 4.—RIEDLER TURBINE WITH FOUR WHEELS.

Figure 4 represents a Riedler and Stumpf turbine of 500 kilowatts, in four successive chutes of steam.

With the exception of turbines of sufficiently small powers to allow the use of speed-reducing gearing, the application of turbines of one wheel is not considered practicable, except for pumping engines having centrifugal pumps, and for ventilating engines, both delivering at a great height, the utilization in these cases being immediate at extremely great rotary speeds.

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Thus, the turbines with multiple-wheels are the only ones of a truly general application.

TURBINES WITH MULTIPLE-WHEELS.

DRUM-TURBINES WITH ROTATING CIRCLET OF BUCKETS.

The characteristic of this kind of turbine, is the employment of a rotating drum carrying a circlet of buckets similar to each other.

The ensemble of a turbine thus constituted consists of several drums of different diameters, each surmounted by a certain number of circlets of buckets.

The prototype of the drum-turbine is the turbine of Parsons.

The Parsons turbine of 1,500 horsepowers furnished to the City of Elberfield, and with which experiments giving astonishing results have been made, is represented in Figure 5, Plate II.

This turbine operates an alternator of 1,000 kilowatts, revolving with the speed of 1,500 revolutions per minute.

The construction of the rotating band of buckets and of the distributor, as practiced by Parsons, is represented in Figure 6, Plate II.

With the exception of dimensions, this kind of construction is generalized for this variety of turbine.

The distributor is formed of a band of curved channels fixed in an exterior envelope. This band is keyed to the drum. The height of these channels varies, naturally, with the weight and volume of steam used.

Figure 7, Plate II, shows an interior view of a Parsons turbine, with a rotating drum and the successive distributors fixed into the casing, removed.

Figure 5 shows a vertical section of a Parsons turbine, from which its principle of distribution can be easily understood. In the large turbines, the rotating drum carries several different diameters of bands of buckets. To equilibrate the longitudinal push or axial thrust, the drum is sometimes constructed symmetrically relatively to a medium plane perpendicular to the axis, but the number of rotating bands of buckets must be

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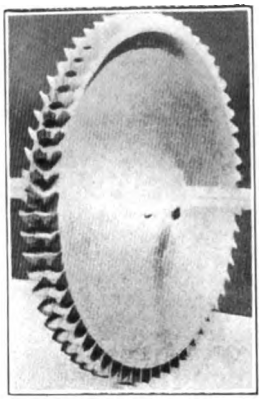

Fig. 1.—Rateau Helicoidal Wheel.

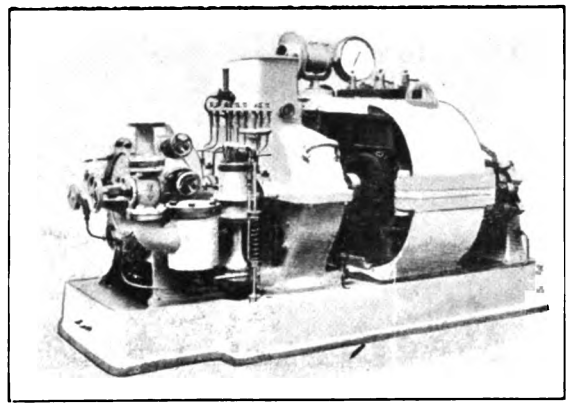

Fig. 2.—Turbine of One Wheel, Rateau System.

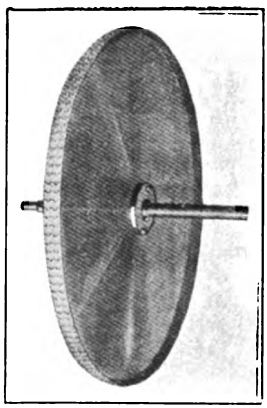

Fig. 3.—Riedler-Stumpf Wheel.

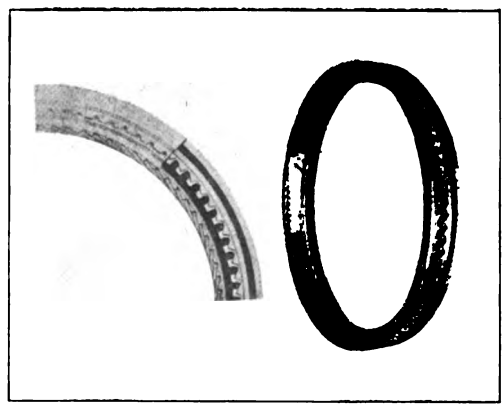

Fig. 4.—Riedler-Stumpf Distributors.

PLATE I.

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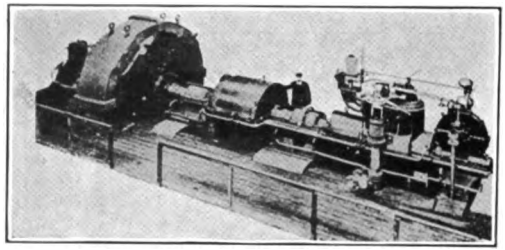

Fig. 5.—Parsons Turine, Elberfeld,

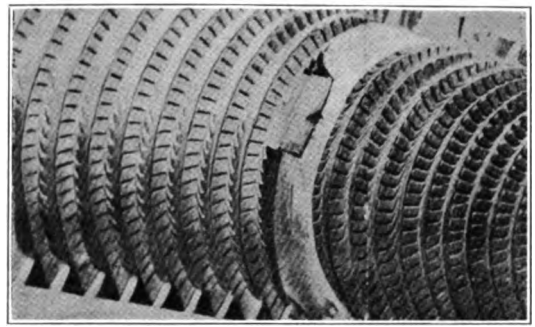

Fig. 7.—Interior View of Parsons Turbine.

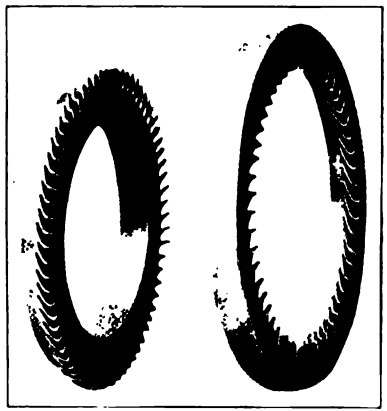

Fig. 6.—Rotating Band of Buckets and Distributor, Parsons.

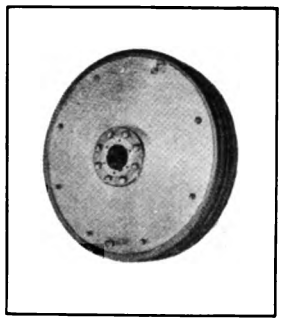

Fig. 8.—Three-Wheel, Curtis.

PLATE II.

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doubled in that case. For all the wheels the distribution is total, that is to say, the rotating bands of buckets are fed with steam over their entire periphery, which is a necessity when the steam acts reactively, as will be shown.

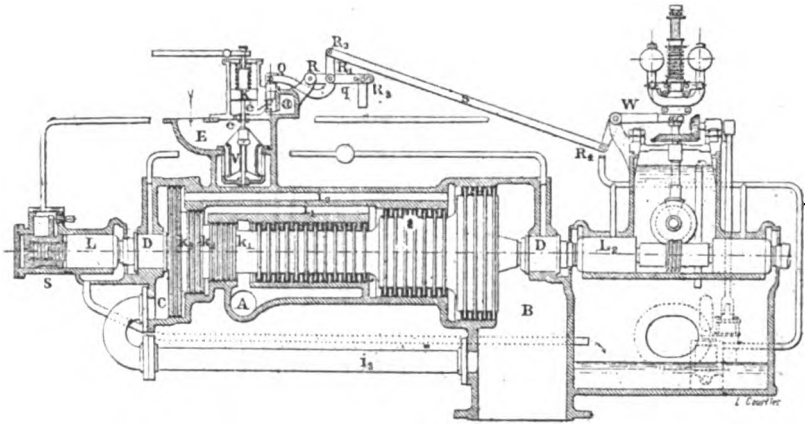

Fig. 5.—Vertical Section of a Parsons Turbine.

TURBINES WITH GROUPS OF WHEELS.

Instead of expanding the steam successively in each rotating wheel or moving band of buckets, the expansion can be effected by groups of wheels, each group using one expansion, or one fall or chute of steam, the speed of efflux created by the fall being maximum on the first wheel of the group while the succeeding wheels of the group only utilize the speed remaining at the outflow from this first wheel.

This principle of distribution was designed for the first time, it is believed, by Mr. Mortier under the name of distribution by stages of speed, as opposed to the distribution by stages of pressure as in the turbines of Parsons and of Rateau. This system could also be termed the turbine system with falls of speed.

Among the machines actually based on this principle, and which have already been constructed, is the American Curtis turbine. Some examples of this turbine will be given in order that its mode of construction may be understood.

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Figure 6 represents a combination of a Curtis turbine with an alternator. It has two groups of three wheels each and is of 500 kilowatts power.

The ensemble of the Curtis turbine composed of three rotating wheels, is shown in Figure 8, Plate II.

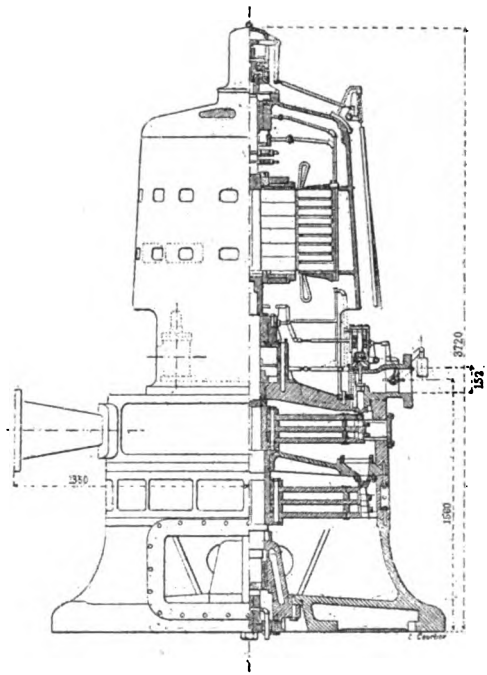

Fig. 6.—CURTIS ELECTRIC GENERATING SET WITH TWO GROUPS OF THREE WHEELS.

As will be seen, the first wheel utilizes the complete expansion of the steam, so that there results by means of the first distributor the full fall of the pressure of the steam admitted.

Between the first and the second wheel is a second distributor whose only effect is to utilize the speed remaining in the steam when it leaves the first wheel, and to thus send that steam into the second wheel with the proper direction.

A third distributor is placed on the exhaust side of the second wheel to utilize the speed there remaining in the steam,

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but much less than at the exhaust of the first wheel; it directs the flow of steam to the third wheel.

Figure 7 represents one of the latest groups of electrogens and Curtis turbines; it is described in the issue of "Engineering" of the 5th February, 1904, Figure 2. The two groups of three wheels each, are replaced by four groups of only two wheels each.

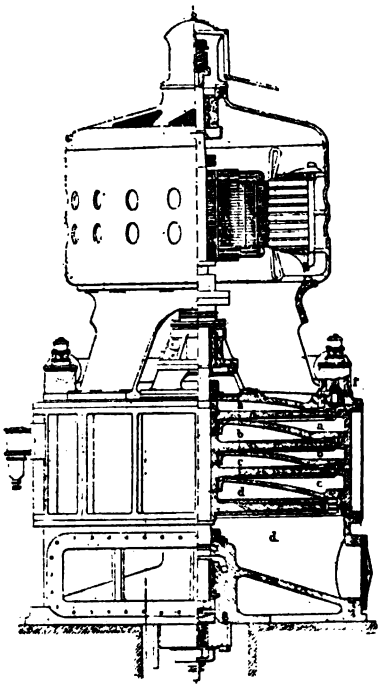

Fig. 7.—Curtis Electric Generating Set with Four Groups of Two Wheels.

According to the article referred to, this ensemble ought to furnish a power of 2,500 kilowatts at the speed of 750 revolutions per minute.

Observe that the Curtis turbine utilizing the steam by impulsion, as in the case of the Rateau turbine, will be identical with the latter when the number of wheels per group is reduced to unity.

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In Figure 9, Plate III, are represented parts of the bucketing of a Curtis wheel for a power of 1,500 kilowatts. The buckets are dovetailed into the wheel and furnished with a tenon supporting a felly at the periphery.

MULTICELLULAR TURBINES.

The sole multicellular turbine actually in use is that of Rateau.

Its essential principle consists in employing the steam in different pressure stages; each wheel utilizes a certain determined fall of pressure, which is a fraction of the total fall from the pressure in the boiler to the pressure in the condenser.

Each wheel works in an isolated manner; it is enclosed in a small cell or apartment formed by two fixed diaphragms extending from the shaft to and into the casing.

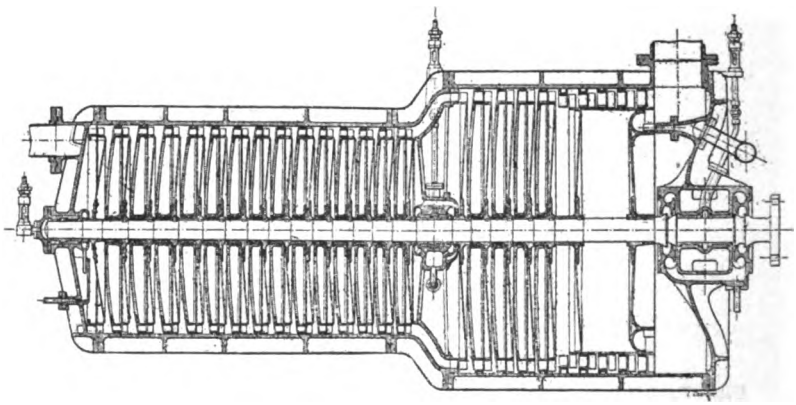

Fig. 8.—VERTICAL SECTION OF RATEAU TURBINE.

The vertical section, Figure 8, of a Rateau turbine, shows clearly this arrangement. The figure represents an old design with inside pillow-blocks, and was calculated for 1,200 horsepowers at 1,800 revolutions per minute.

The wheels used in the Rateau turbine are of plate iron.

At their periphery, fixed upon the felly, are the buckets provided for each band.

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One of these wheels is represented in Figure 10, Plate III.

The distributors of the Rateau wheels are formed of curved buckets held between cheek pieces having the same curvature.

The distribution is made according to the principle of partial injection of steam, up to when the volume of steam required becomes such that there is necessity to inject it around the entire periphery of the wheel; in this latter case, the distributor occupies no longer an arc of the circle or divers arcs of the circle, but the complete circumference of the circle.

The distributors are fixed in logements or chambers arranged on the periphery of those discs which form diaphragms, and which constitute the front and the back of each of the wheels.

The diaphragms are constructed like a wheel with hub, spokes, and felly, covered over with an exactly fitting plate.

Figure 11, Plate III, shows a series of Rateau diaphragms with distributors to the periphery. These diaphragms are in a single piece and are of different diameters.

There can be employed either diaphragms in one piece, or diaphragms in two pieces.

The turbine whose section is given in Figure 8, is constructed with diaphragms in two pieces.

A view of that turbine, which has been the model chosen by the French Marine for the torpedo boat *No. 243*, is given in Figure 12, Plate III.

There will be remarked the dismounting facilities of this kind of construction.

Figure 13, Plate III, represents an ensemble of wheels and of distributors of a multicellular turbine with diaphragm in a single piece. On the shaft are keyed the different wheels and between them are placed the distributing diaphragms.

It is important to know that the clearance which could cause leakage of steam, leads down to the periphery of the shaft and into the annular space left free by the clearance for the hub of the diaphragm. This clearance of about 0.04 inch corresponds to a minimum space, and the proportion of steam which could escape from one wheel to the other without being utilized does not exceed in practice some hundredths.

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There should also be remarked the holes pierced in the plate iron to avoid all longitudinal reaction.

Each wheel then moves in a cellule wherein the pressure is equal on all the surfaces.

Figure 14, Plate IV, represents a turbine of one body of 500 horsepowers measured on the shaft. The arrangement of the pillow-blocks should be remarked: they are outside of the body of the turbine, as well as the command of the regulator of the speed, which latter is supported on the principal pillow-block.

The body of this turbine contains two series of wheels of different diameters.

In Figure 15, Plate IV, is seen a turbine of two distinct bodies of 700 horsepowers on one shaft. It is necessary when the power surpasses a certain limit to fraction or divide the body of the turbine: there are experienced, in fact, great difficulties in keying on the same shaft too great a number of rotating wheels because the distance between the pillow-blocks becomes too large.

Figure 16, Plate IV, represents a marine turbine of two bodies, placed on two different shafts and combined in one connected series. The small body is termed "high pressure," and the other is termed "low pressure."

This turbine of about 2,000 horsepowers has been recently applied to the propulsion of a British torpedo boat.

Figure 17, Plate IV, shows a mixed type, high and low pressure and two bodies.

The large body called the "low pressure" can be separately supplied with the exhaust steam of a pumping engine pumping out a mine.

The "high pressure" body is supplied by two sources of steam, the one of 170 pounds pressure per square inch, the other of 70 pounds pressure per square inch, coming from different boilers.

As long as the exhaust steam is in sufficient quantity to do the full work of the machine, the "high pressure" body does not receive steam and an automatic connection puts it in com-

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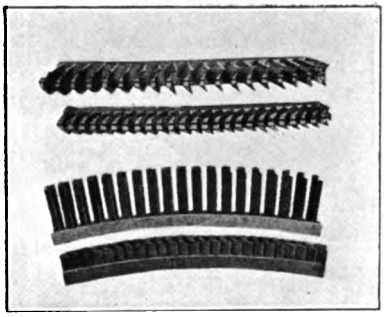

Fig. 9.—Parts of Bucketing, Curtis Turbine.

Fig. 10.—Rateau Wheel of Sheet Iron.

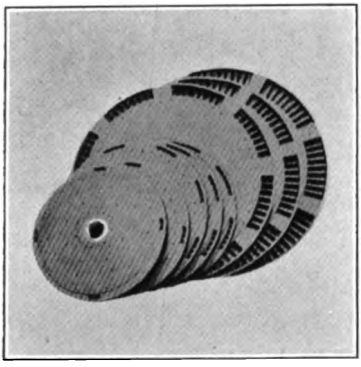

Fig. 11.—Series of Rateau Diaphragms.

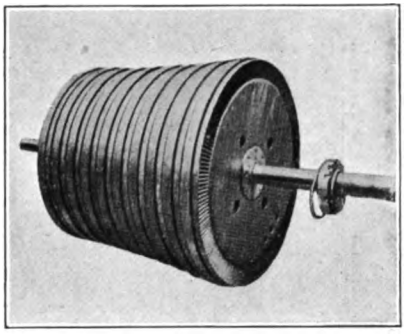

Fig. 13.—Ensemble of Wheels and Distributors, Rateau.

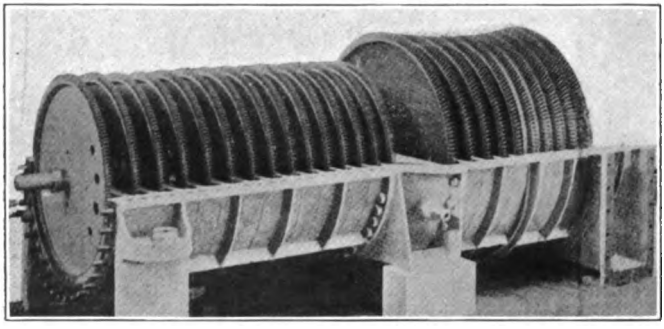

Fig. 12.—View of Turbine of Torpedo Boat 243.

PLATE III.

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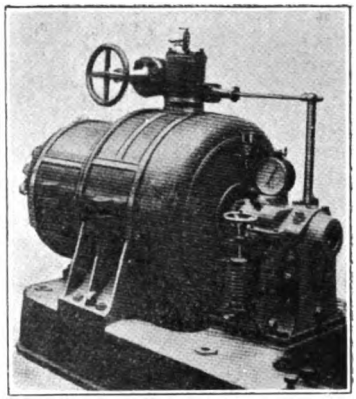

Fig. 14.—Turbine of one Body, 500 Horsepowers.

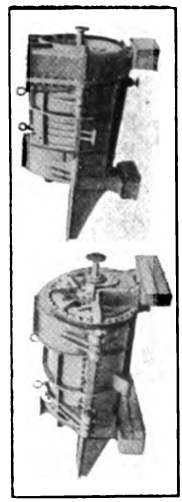

Fig. 16.—Marine Turbine of Two Bodies.

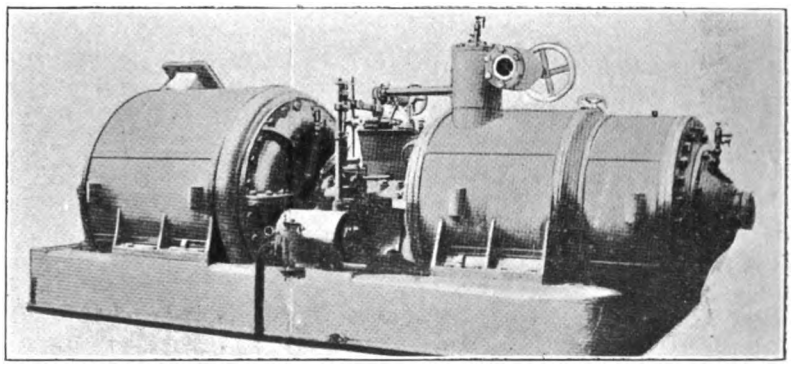

Fig. 15.—Turbine of Two Bodies, 700 Horsepowers.

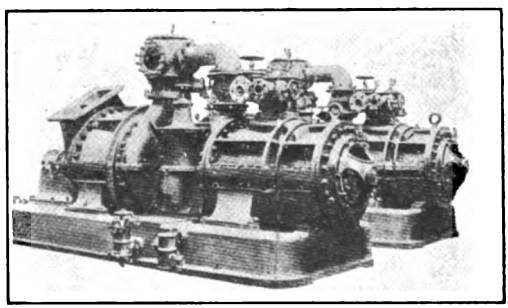

Fig. 17.—Mixed Type, High and Low Pressure.

PLATE IV.

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munication with the condenser, so as to reduce the friction of the wheels on the steam to a minimum.

When the exhaust steam no longer suffices for doing the work, the arrival of steam takes place in the "high pressure" body, and the steam can thus expand in the ensemble of the two bodies.

There can consequently be utilized by this arrangement steam produced by or coming from, three different sources. It is superfluous to insist on the advantages of such a combination which appears to be quite unrealizable in a piston-engine.

II.—CLASSIFICATION ACCORDING TO THE ROUTE FOLLOWED BY THE FLUID-MOTOR.

The route followed by the Steam in passing through the Turbine, may be in either of two directions.

The steam can follow a trajectory parallel to the axis, or a trajectory normal to the axis.

In the first case the turbine is termed "axial" or "helicoidal"; in the second case it is termed radial.

AXIAL OR HELICOIDAL TURBINES.

Up to the present time, the only turbines employed in practice are of the axial variety.

It will suffice to pass only them in review.

In the Parsons turbine the fluid moves parallel to the axis. It enters either at one end of the turbine to flow out of the other end; or it enters into the diametral plan of the turbine and flows towards the two extremities.

In the Rateau turbine, the fluid follows a route also parallel to the axis; it enters at one extremity and escapes at the other.

In the Laval turbine, the steam enters by one of the faces of the wheel and leaves by the other. It is the same as in the first Rateau turbine, or in the Riedler-Stumpf turbine, the trajectory having the well known form of that of the Pelton wheel; the fluid enters by the diametral plane of the wheel, and leaves by the two faces of the wheel, but its direction is always parallel to the axis of the wheel. The Curtis and Westinghouse turbines are likewise axial turbines.

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RADIAL TURBINES.

The Radial Turbines are divided into two groups:

Centrifugal Radial Turbines,

Centripetal Radial Turbines.

In the first, the fluid flows from the axis of the wheel to its periphery.

In the second group, the fluid, on the contrary, enters at the periphery and leaves at the axis of the wheel.

CENTRIFUGAL RADIAL TURBINES.

There can be cited as the first, the Centrifugal Radial Turbine of Dow, composed of a series of rotating concentric circlets of buckets into which the fluid is delivered successively from a corresponding series of directrices.

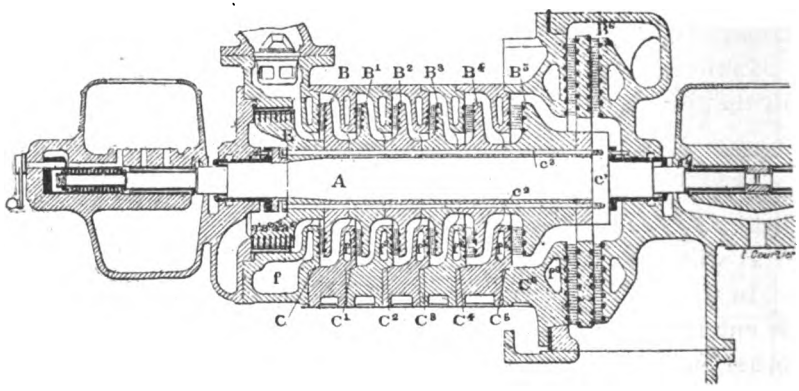

Fig. 9.—PARSONS' CENTRIFUGAL TURBINE.

The Multicellular Centrifugal Radial Turbine of Edwards, in which the steam traverses the distributing canals or passages directed from the axis to the periphery, and acts on rotating buckets fixed to the body of the wheel.

At the extremity of the turbine, the expanded steam acts in an inverse manner to the centripetal flow.

Parsons has also exercised his ingenuity on the construction of a centrifugal radial turbine represented in Figure 9.

The construction is somewhat analogous to that of Dow's

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turbine. The rotating circlets of buckets are concentric, and at the periphery of each series the flow of steam is led into the axis of the following series by an oblique passage or canal.

CENTRIPETAL RADIAL TURBINES.

One of Parsons' patents shows a Centripetal Radial Turbine (Figure 10); the construction is the inverse of the preceding turbines.

Dumoulin, also, has invented a Centripetal Turbine of which a description is given in Sosnowski's work.*

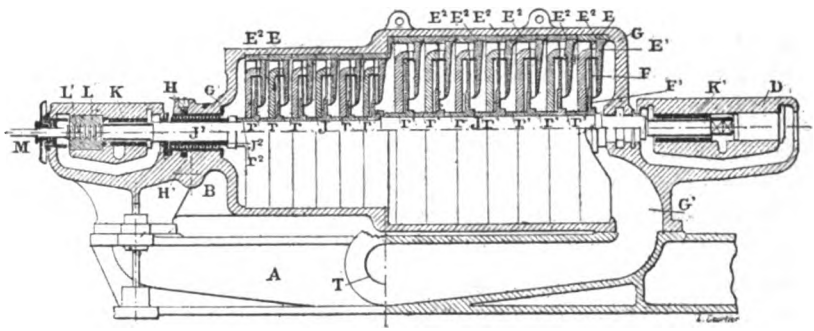

Fig. 10.—PARSONS' CENTRIPETAL TURBINE.

Finally, in the same Memoire will be found the description of Last's original machine, in which each wheel works centrifugally and centripetally; the flow of the steam is always slightly oblique to the axis; so that this machine is simultaneously a heliocentrifugal and a heliocentripetal turbine.

III.—CLASSIFICATION ACCORDING TO THE MODE IN WHICH THE MOTOR-FLUID FUNCTIONS.

The steam can act in the Turbine according to two different Modes.

It can function by action or impulsion, and by reaction.

The steam, in its mode of action, completely expands in the distributor, the fall of pressure being wholly employed to create all the speed of which the steam is susceptible; the

*Roues et Turbines à Vapeur. Paris. 1901.

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energy derived from the expansion is found integrally in the *vis viva* of the molecules of the steam thus put in movement.

The push on the movable buckets is then wholly due to the quantity of movement of the steam, this push being rigorously equal to the product of the mass into the speed, taking into account, in the calculation, the effect of the variable inclination of the steam on the rotating buckets during its action; the density remaining invariable during the route across the buckets, the cross section of the route can remain constant.

In the method of reaction, the expansion of the steam is only partially accomplished in the distributor, and is completed during the trajectory of the steam in the passages or canals of the moving buckets.

The speed created by the outflow from the distributor, for a given fall of pressure, is then less than in the preceding case, the fall of pressure between the entry into, and the exit from, the distributor being very much less.

Consequently, the steam acts in the rotating buckets simultaneously by its quantity of movement and by its pressure, in measure as the expansion takes place in the rotating buckets.

The speed of circulation increases, and the cross section of the passage ought not to remain constant. When the steam does work according to this mode of action, the two sides or faces of the rotating wheel are subjected to the same pressure, since it rotates in an entirely expanded fluid.

On the contrary, in the reaction turbine, the two sides of of the rotating wheel are subjected to different pressures, and this difference is so much the greater as the degree of reaction is, itself, greater.

LAW OF EFFLUX OF SATURATED STEAM.

The mode of functioning of the fluid-motor depends essentially on the form and on the direction of the buckets, passages, or channels, in which it moves, be they fixed or rotating.

The form of these buckets, passages, or channels, is determined by the law of efflux of saturated steam.

The efflux of elastic fluids by means of rectangular or curved

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distributors, differs much from the efflux of liquids for the cases in which the proportion of the higher pressure to the lower pressure producing the flow is notably less than unity.

Calling I the weight of the discharge per second, S the area of the cross section of the stream of discharge, δ the specific weight of the fluid in that cross section, and V the velocity per second of the discharge, these quantities have to each other the simple relation

$$I = SV\delta,$$

which is the equation of continuity.

If the specific weight δ be constant, as in the case of liquids, S is inversely proportional to V, and as the velocity V increases always in the same measure that the pressure decreases, S continuously diminishes. Thus, the distributing canals of hydraulic turbines are always made convergent. With elastic fluids, this is not the case; with steam, the specific weight decreases in measure as the pressure decreases during the increase of velocity, in such a manner that the product, which at first was continuously increasing, passes by a maximum and becomes continuously decreasing.

In the case of a gas, the maximum occurs for the proportion

$$\frac{\eta}{P} = 0.52;$$

with steam, this proportion is about 0.58, whence results that when the lower pressure η is lower than 0.58 of P, the tuyere of efflux, at first converging, should then become diverging, if the steam is to be expanded still more, in order that it may obtain all the velocity possible for its fall of pressure from P to η. The proportion of the area of the final cross section of the tuyere to the cross section at the section S of the neck, should vary with the proportion $\frac{\eta}{P}$ of the pressures, while the pressure at the neck is always equal to 0.58 of P.

The quantity of steam discharged is independent of the low pressure η when that pressure is less than 0.58 of the high pressure P; but, on the contrary, the quantity of steam dis-

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charged depends also on the pressure η when that pressure is greater than 0.58 of the pressure P.

There are then two very different cases to be distinguished:

In the first case, the calculation of the quantity of discharge depends only on P, and the formula is simple. It can be expressed as follows, as shown by Mr. Rateau:

$$I = S P (15.20 - 0.96 \log P).$$

In the second case, the calculation of the quantity of discharge depends simultaneously on P and on η.

In both cases, the smallest section of the tuyere (at the neck, if the tuyere be convergent-divergent, or, at the discharging end, if it be only convergent) must be taken for the calculation of the quantity of discharge.

Thus, when the lower pressure η is greater than 0.58 of P, which is the general case of steam-turbines with multiple wheels, convergent tuyeres must be employed.

On the contrary, when η is smaller than 0.58 of P, which is the case of steam-turbines with one wheel, like Laval's turbine, convergent-divergent tuyeres must be employed if the complete expansion of the steam in the tuyere be desired in order that the expanded steam may have all the velocity which corresponds to its fall of pressure from P to η.

Figure 11 shows the curve giving the values of the successive diameters of a conical tuyere in function of proportion of expansion. The abscissae show the ratio of expansion of the lower pressure η to the higher pressure P, and the corresponding ordinates show the values of the diameters.

When the ratio of the lower pressure to the higher pressure falls below 0.58, the tuyere becomes convergent-divergent; the divergent part has unfortunately an injurious influence on the economy. As Mr. Rateau has already observed in his turbines, and as Professor Stodola has experimentally demonstrated, the stream of steam has a tendency to separate from, or to leave the surface of, the tuyere in the divergent part, thus producing a back pressure against the steam-current at the moment of this separation.

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To avoid, at least partially, this defect, an angle of small conicity, not surpassing 3 or 4 degrees, must be given to the divergent part of the tuyere. Under these conditions, when the ratio of the expansion is considerable, the convergent-divergent tuyere should have a great length, which can be particularly inconvenient when the discharge of steam is considerable.

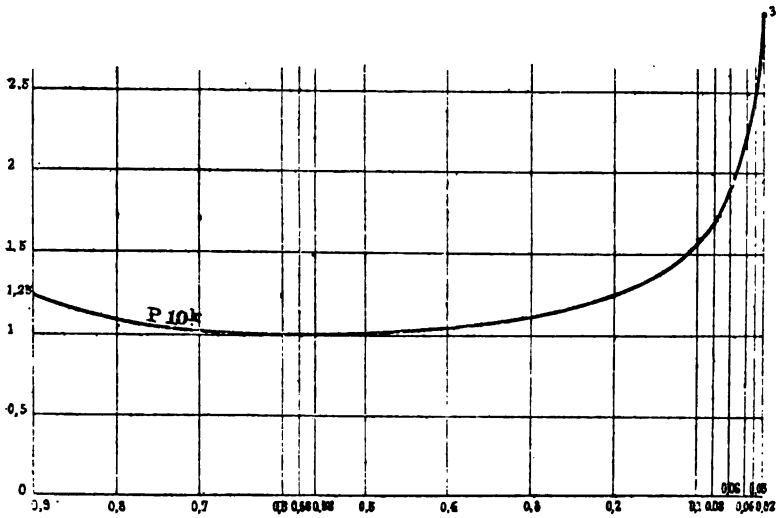

Fig. 11.—Profile of the Tuyeres for a Steam Turbine.

Supposing a minimum section of unit diameter, and a discharging section of treble this diameter, then, that the total angle of divergence remain at 4 degrees, the length of the tuyere must be equal to 29 times the diameter of the minimum section.

With a tuyere discharging, say 441 pounds weight of steam per hour, and with a pressure P of 147¼ pounds per square inch above the absolute zero, the minimum section would have a diameter of 0.028 inch, the length of the divergent will be 0.812 inch, the expansion being only 2 per centum, which corresponds to the lower pressure η of 2.94 pounds per square inch.

The necessity of giving a great length to the converging-

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diverging tubes, compels their number to be increased in order to reduce their diameter, and, in fact, all their dimensions. It is a serious matter with steam-turbines of one wheel, but it does not exist for steam-turbines of multiple wheels, in which the tuyeres are all convergent.

COMPARISON OF THE BUCKETING OF A TURBINE OF ACTION AND OF A TURBINE OF REACTION.

Figure 12 represents the Bucketing and Steam-passages, fixed and mobile, of a Turbine of Reaction with multiple wheels, and those of a Turbine of Action.

The form of the passages in fixed or stationary distributors is the same in both cases.

On the contrary, the form of the Buckets in the revolving wheels is notably different.

In the reaction turbine, the Buckets of the revolving wheels have a section decreasing from the entrance of the fluid to the leaving of the same. This fact is due to the expansion itself of the steam whose velocity increases from the entrance to the discharge; the section of the buckets follows the law just shown. To realize this law, the bucket must be slightly increased at its center. All experiments have shown that the mobile buckets can be constructed of plate iron of equal thickness and of circular form placed symmetrically to the diametrical plane of the axis: the section of the bucket is not then rigorously constant, but the difference from the theoretical section is slight and the economy is not sensibly affected.

To distinguish, at a glance, the wheel of a turbine of reaction from the wheel of a turbine of action is easy.

In the first, the buckets are not symmetrical in relation to the diametrical plane; and, in the second, they are symmetrical.

The same figure, 12, gives the diagram showing the velocities in a turbine of action.

The absolute efflux of the steam at the entrance, is given with the angle of inclination of the distributor.

This speed is compounded with two other speeds:

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The speed of entrainment of the steam, or peripheral speed of the wheel, and the relative speed of the steam in the buckets; this is inclined from the angle itself of the bucket at the entrance.

The relative speed of the steam at its discharge is naturally less than its relative speed when entering. This reduction is due to frictions and eddies of the gaseous steam during its trajectory through the passages.

By compounding this last relative speed with the speed of entrainment, there will be obtained the absolute speed of the steam at the moment of its discharge, and its direction.

From the diagram of the speeds, the hydraulic economy of a turbine of action can easily be calculated.

CONSEQUENCES OF THE MODE IN WHICH THE FLUID-MOTOR FUNCTIONS IN TURBINES.

The mode in which the steam functions, whether by impulsion or by reaction, has very important consequences for these two categories of turbines.

In the turbine of reaction, it is necessary:

1st. To equilibrate the axial push or thrust produced by the difference in the pressure of the steam upon the two faces of each revolving wheel.

2d. To allow but a very slight clearance between the stationary and the moving parts in order to prevent steam-leakage from one face to the other of the wheels.

3d. To admit steam simultaneously to the entire periphery of the wheel, in order to avoid loss by eddies when the canals of the moving wheels arrive opposite to those of the stationary wheels.

The schematic design in Figure 13, shows the fundamental differences which separate the functioning of a drum turbine of reaction in the Parsons system, from that of a multicellular turbine of action, both turbines being helicoidal.

The mode of action causes a strong difference of pressure on the two faces of the bands carrying the buckets; and, further, the total difference of pressure between the extreme

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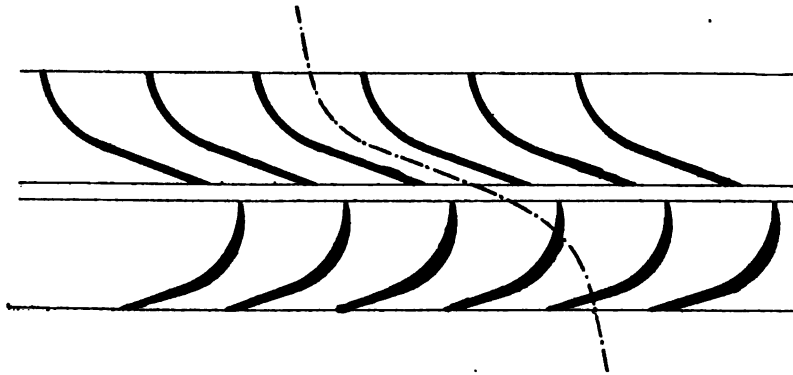
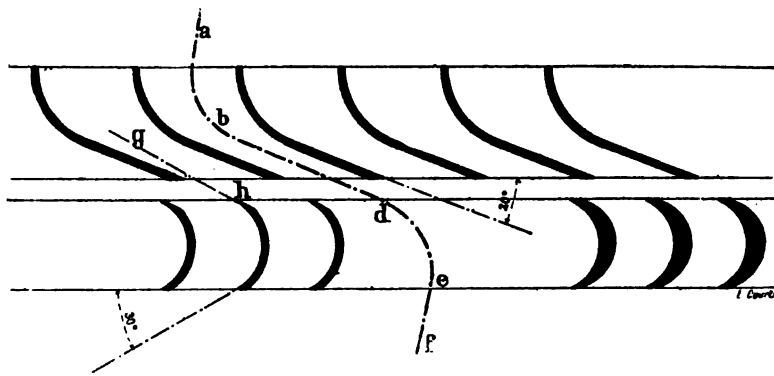
pressures P and η, acts on the two exterior faces of the drum carrying the buckets.



TURBINE OF REACTION WITH MULTIPLE WHEELS.



TURBINE OF ACTION WITH MULTIPLE WHEELS.

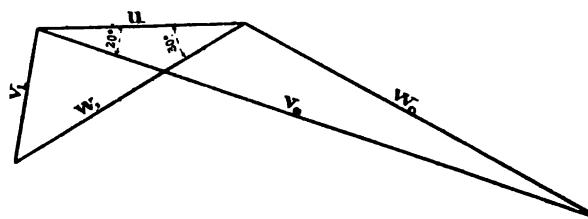


DIAGRAM OF SPEED FOR A TURBINE OF ACTION.

Fig. 12.

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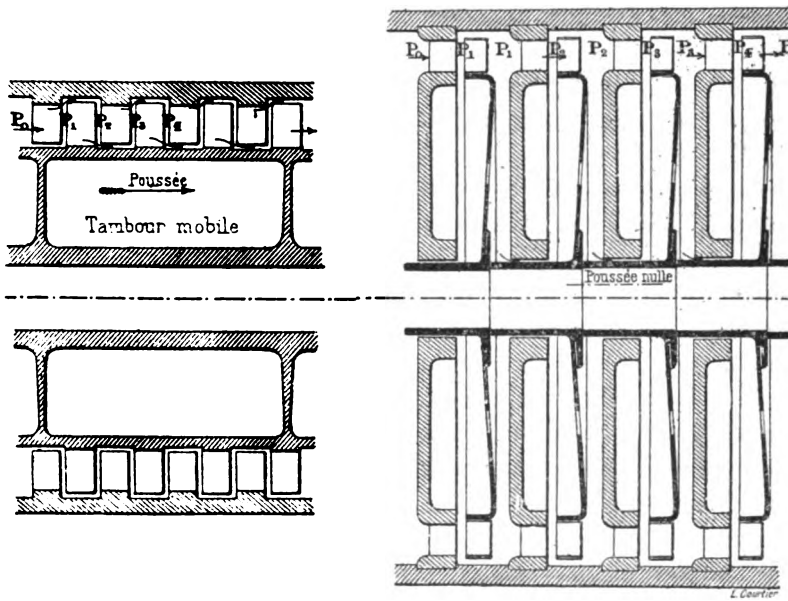
This axial pressure, which, for large turbines, may amount to several thousands of pounds, has to be equilibrated.

In the Parsons system, this pressure is equilibrated by making the steam pass simultaneously in two opposite trajectories; or, by adding an equilibrium cylinder with movable piston to the extremity of the turbine, steam being admitted to the two sides of the piston and respectively of the two pressures P and η.



TURBINE OF REACTION WITH MULTIPLE WHEELS, PARSONS SYSTEM.

MULTICELLULAR TURBINE OF ACTION, RATEAU SYSTEM.

Fig. 13.—PATH OF STEAM AND AXIAL THRUST IN HELICOIDAL TURBINES.

In a multicellular turbine of action there is no unbalanced axial pressure to be counterbalanced.

In the reaction turbine, the clearance between the rotating wheels and the envelope casing, or shell, should be reduced to a minimum. And the same may be said of the clearance between the exterior periphery of the drum and the internal face of the fixed band of distributors.

In the multicellular turbine, the clearance can be consider-

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able, and attains, in practice, as much as from 0.12 to 0.16 inch between the fixed part, distributor or envelope, and the rotating wheel. Only the clearance between the shaft and the internal hub of the distributor should be reduced to about 0.04 inch.

When the movable buckets of a reaction turbine, full of steam, come by its movement in front of a fixed or blank side, which will happen if the distributor extends over only an arc of the periphery, there is produced by the fact of the pressure in the buckets a reflex of steam in a direction opposite to the direction of the movement, causing strong frictions and considerable eddies that absorb a part of the energy.

This phenomenon compels, in the case of reaction turbines, a distribution over the whole of the periphery; also it compels the first wheels on the side of the high pressure to be of small diameter, because, for high-pressure steam, if large wheels be employed, the aggregate sections of the distributor would lead to partitions of too small a height, not exceeding 0.04 to 0.08 inch.

The hydraulic economy of small diameter wheels being very much less than that of wheels of large diameter, it is necessary to notably augment in reaction turbines the number of wheels which thus becomes more than double the number required in turbines of action.

The remark may not be superfluous that the mode of functioning by action when employed in a multicellular turbine has very serious advantages, both as regards the construction and as regards the simplification of the machine.

REMARKS ON THE CLASSIFICATION OF STEAM-TURBINES.

The conclusion that clearly results from the general classification of steam-turbines, and which the examination of the economy given by them confirms, is that all the various systems designed merge in two forms, namely, the drum reaction turbine with multiple bands of bucketing, devised by Parsons; and the multicellular turbine of action devised by Rateau.

By combining two or several of the systems of each category,

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new systems can be formed participating in the properties of the different systems selected. In this manner, for example, a drum turbine of reaction can be combined with a multicellular turbine of action.

And, still further, by taking on the one part the wheels of the Pelton system with tuyere distributors as first employed by Rateau and later by Stumpf; and, on the other part, by taking the helicoidal wheels as afterwards employed by Rateau; to constitute a multicellular turbine of action of a special system.

But all these hybrid systems reunite the disadvantages of their different prototypes composing them, without having all of their advantages.

Although the drum turbine of action with bands of moving buckets, compared with the drum turbine of reaction, permits, like all turbines of action, partial injection of the steam and a larger clearance between the buckets and their casing or envelope, it restricts, on the contrary, to a smaller clearance between the drum and the internal periphery of the distribution for preventing the internal leakage of steam.

These leaks are greater than in the Parsons turbine, because the expansion being complete in each distributor, the difference of pressure between the two faces will be a maximum.

In the Rateau multicellular turbine, the clearance in question is carried down to the periphery of the shaft, and the leakage is proportionally reduced.

The drum turbine of action participates, it is true, in the property of the turbines of action to avoid all axial thrust on the bands of moving buckets, but they have that thrust on the drum itself, the faces of which are subjected to the difference of pressure between the maximum and the minimum pressure in the turbine.

It will then be perceived that this type, though based on the principle of action, does not benefit by all of the advantages of that principle, and it retains several disadvantages of the principle of reaction.

The mixed multicellular turbine, having Pelton wheels and

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helicoidal wheels, has the disadvantage of a mediocre economy for the wheels of the first kind under penalty of either multiplying their number and of rendering the construction costly, or of too much increasing their diameter.

The velocity remaining in the steam when leaving the Pelton wheels is absolutely non-usable, which still more contributes to lessen the economy. Also, there can easily be foreseen that this mixed system will, by the suppression of the Pelton wheel, fatally end in the Rateau multicellular turbine, and that the preceding mixed system will end in the Parsons turbine.

The final experimental result, then, and to which the logic of physical and mechanical laws also leads, is that there can be only two fundamental forms of steam-turbines, namely, that of Parsons and that of Rateau, of which all others are but variants.

ECONOMY [RENDEMENT] OF STEAM-TURBINES.

DEFINITION OF THIS ECONOMY.

The economy of a steam-engine is one of the most universally accepted notions among engineers, and, at first view, seems one of the most precise.

In reality, the fact is very different, and, notwithstanding the works of numerous scientists and experimenters, nothing can be less definite than the economy of a steam-motor, taking the term at hazard as employed by any constructor whomsoever.

Even yet, for a great many persons, the economy is measured by the weight of fuel consumed by any given engine per unit of work done, the definition of which is often of the most vague kind.

This mode of measurement which involves in it the economy of the boiler and of the auxiliary apparatus, can be still more vitiated by the manner of considering what is coal; whether the ash, clinker and hygroscopic water that it may contain be or be not included in the term, and all of which is very vari-

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able for different coals, as well as the calorific power of different coals, which is quite as variable.

As for the work done by the motor, it was for a long while measured by the dimensions of the cylinder which was supposed capable of producing a given power at a determined speed.

Later, the discovery was made that a more certain method was necessary, and then the weight of steam actually used was substituted for the weight of coal consumed.

When the problem concerns engines of great power, the determination of the weight of steam used is a much more delicate operation than it seems at first sight.

As regards the measurement of the useful power developed upon the shaft, it is always difficult when the power exceeds 40 or 50 horsepowers, unless the motor is directly attached to a generator of electricity.

The only process in general use—indicator diagrams taken from the cylinder—does not in most cases give the economy with sufficient precision.

The diagram gives well enough the internal or indicated work done, but as the proportion between the effective work really done on the shaft and the indicated work is always unknown unless the effective work can be directly measured, the information furnished by the indicator must necessarily be received with caution.

The proportion between the effective work and the indicated work, varies, according to authors, from 75 to 93 per centum: 90 per centum is generally admitted for a steam-engine in good condition, but a rigorous demonstration cannot be given, as that requires the use in each case of a dynamo performing the rôle of a dynamometer.

Only recently have the processes employed in electrical technology been applied to measure with precision the work done upon the shaft of a generator of electricity by its steam-motor.

This method of measuring is nearer exact than any of the other methods attempted; measuring by the brake being im-

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possible when the power developed by the engine, and its rotary speed, surpasses certain limits.

Once the number of effective horsepowers developed by a steam-engine, and its corresponding consumption of steam, being known, what is intended by its economy [rendement] must be exactly defined.

The economy of a steam-turbine is the thermal economy represented by the relation between the effective energy furnished to the shaft and the total disposable energy contained in the corresponding weight of steam used per unit of time; the fall of pressure which renders this energy disposable, is equal to the difference between the pressures of the steam entering and leaving the machine.

To calculate the energy of a given machine, the effective power of which and its consumption of steam has been ascertained, the total energy disposable under the conditions of its use must be known.

CALCULATION OF THE TOTAL DISPOSABLE POWER CONTAINED IN THE STEAM.

The calculation of the total disposable power contained in a pound weight of steam expanding from its higher pressure P to its lower pressure η, is given by the laws of thermodynamics and by the experiments of Regnault.

This calculation can be made either by the aid of formulas, or by the aid of the entropic diagram the use of which is daily becoming more and more extended.

During the investigations of the steam-turbine made by the author in 1896, he constructed a table that gave for any fall of pressure, whatever, the disposable energy of one pound weight of steam.

This table having been calculated, was sent to Mr. Rateau who conceived the ingenious idea of translating it under the form of an abacus with a logarithmic base, reuniting on one line the items corresponding to an equal consumption of steam.

This abacus is represented in Figure 14; the abscissas are

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the absolute high pressures P, and the ordinates are the absolute low pressures η.

In constructing this abacus, Mr. Rateau soon perceived that the lines of equal consumption of steam were not curves as is generally the case with elastic fluids, but straight lines. The coincidence was so perfect that he was enabled to immediately deduce an empirical law for the calculation of the theoretical

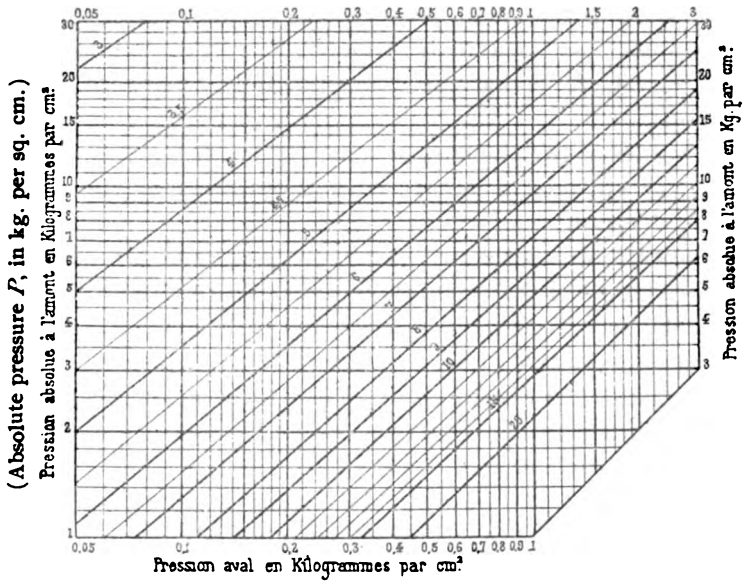

Fig. 14.—ABACUS OF THEORETICAL CONSUMPTION FOR A STEAM-ENGINE.

consumption of steam per horsepower per hour in pounds weight of steam evolved between two given pressures. This empirical law is remarkably in accord with the results drawn from the tables of Regnault; the differences are about the one thousandth part.*

Knowing, by means of Mr. Rateau's abacus, the theoretical consumption K represented by the effective horsepower per

Abacus of the theoretical consumptions by a steam-engine, and new laws relative to steam, by Mr. Rateau. Extract from Annales des Mines, issue of February, 1897.

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hour in the given conditions, the speed of outflow of the saturated steam between the same high and low pressures P and η can be easily deduced. The simplified formula is the following:

$$V_0 = 100 \sqrt{\frac{530}{K}},$$

which gives the speed V_0 of the outflow of the steam in meters [3.2809 feet] per second, between the pressures P and η in kilogrammes per square centimeter, [14.2228 pounds per square inch], K being the corresponding theoretical consumption.

The values of the theoretical consumption furnished by the abacus, are entirely in accord with those deduced from the entropic diagram.

In the case of superheated steam, the abacus of Mr. Rateau can be completed by calculation, aided by the thermic diagram; the superheating augments the utilizable area of the diagram between two given pressures by a supplementary area easy to value. Mr. LeLong, Engineer of the Marine, has furthermore shown an ingenious abacus which enables the calculation to be supplemented.

The abacus of Mr. Rateau, completed when superheated steam is used, allows the valuation to be made of the economy of all steam-engines of which the higher and lower pressures, P and η, are perfectly known and of which the consumption per effective horsepower per hour is known. A precise notion is thus obtained of the economy, which furnishes a truly rational classifying of a steam-engine whatever may be its nature and its mode of construction. The economy should no longer be subordinated to the consumption of steam.

ECONOMY AND CONSUMPTION OF STEAM.

The net economy, and the consumption of steam, are two quite distinct notions, though too often confounded: when the consumption of one machine is given in order to compare it with that of another, the higher and lower pressures P and η

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are not always stated between which the steam is evolved. It can very easily happen that the consumption which appears most advantageous corresponds, in reality, to a lower economy.

The following table shows an example of two steam-turbines, one of which has a small consumption but with the steam received under a high pressure P and expanded down to a feeble pressure η; the other machine, on the contrary, has a large consumption, but then the steam is admitted to it at a notably less pressure, and its vacuum is mediocre.

The table shows that although the consumption of steam by turbine A is 24 per centum less than by turbine B, the economy of the latter is 21 per centum more.

Whence results that were turbine B placed in the conditions of turbine A, its consumption would be 18 per centum less.

TABLE OF THE COMPARATIVE CONSUMPTION OF STEAM BY, AND OF THE ECONOMY OF, TWO TURBINES A AND B UNDER DIFFERENT CONDITIONS.

——	Turbine A.	Turbine B.
Higher pressure P, in pounds per square inch above zero	220.84	147.23
Lower pressure η, in pounds per square inch above zero	1.40	1.16
Superheating added, in degrees Fahrenheit	212.00	0.00
Temperature, in Fahr. degrees, of the steam	566.60	356.80
Consumption of steam, in pounds weight per hour, per effective horsepower	12.12	15.10
Consumption of steam, in pounds weight per hour, per theoretical horsepower	6.46	9.81
Thermic economy	$\frac{6.46}{12.12}=0.534$	$\frac{9.81}{15.10}=0.650$
Ratio of $\frac{B}{A}$	Consumption. 1.246 or 24.5 per ct. in excess.	Economy. 1.217 or 21.5 per ct. in excess.

Turbine B in the conditions of Turbine A. Consumption of steam in pounds weight per hour per effective horsepower $6.46 \div 0.65 = 9.939$ pounds, is 18 per centum less.

HYDRAULIC ECONOMY OF THE WHEEL OF A TURBINE OF ACTION.

The most important term which intervenes in the total economy of a turbine of action, is that which is called the

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hydraulic economy. It is calculated from the diagram of the speeds on entering and on leaving the moving bucketing. This method has been proposed by Mr. Rateau for the calculation of hydraulic turbines. It was applied by him for the first time to steam-turbines in a Memoir on the Elementary Theory of Steam-Turbines, deposited in the Academy of Sciences in 1901.

The calculation of the hydraulic economy depends essentially on the coefficient of the slowing of the relative speed of the fluid in the moving bucketing; this coefficient once known as well as the absolute speed on entering, and the angles of the fixed and moving bucketings, the hydraulic economy can be immediately deduced.

The calculation can be made graphically, as shown by Mr. Rateau, as it is easily translatable into formula.

The hydraulic economy for the given angles of construction of the fixed and moving bucketings, and for a known coefficient of the slowing of the speed, is expressed, in function of the relation of the peripheral speed of the bucketing to the absolute speed of the steam at its entrance; this speed itself being given, the economy is then known in function of the peripheral speed of the wheel.

Both theory and experiment show that the hydraulic economy increases rapidly with the peripheral speed up to a maximum after which it gradually decreases.

In general, the speed corresponding to the hydraulic economic maximum cannot possibly be attained with turbines of multiple wheels, but it can be approximated sufficiently near to realize hydraulic economy up to 70 per centum of the maximum and even surpassing that proportion.

STEAM-LEAKS IN TURBINES OF MULTIPLE WHEELS.

Steam-leakage does not exist in turbines of impulsion with one wheel, because the steam acts only in a single compartment.

In turbines of multicellular action, leakage is only to be feared between the two faces of a distributor; the steam which

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escapes from one face to the other, loses thus its pressure of expansion without any utilization.

There has been previously shown that in the Rateau turbine this defect is almost completely corrected.

In the reaction turbines with multiple wheels, the leakage at the distributor is less, but the leakage around the moving bucketing is considerable; the steam has always a tendency to pass from the one face to the other, since the pressures on these faces are different. The tendency to this leakage is nearly the same from one end of the turbine to the other, while in the multicellular system of action, the leakage around the shaft has a sensible value for only the first wheels where the difference of pressure between the two faces of the distributor is more considerable.

This is another reason that compels increasing the number of wheels in the reaction turbine.

FRICTION OF THE STEAM.

The hydraulic economy is never equal to unity, as has been previously remarked: it tends toward a maximum which it cannot reach in practice.

The difference between the value of the hydraulic economy and unity, constitutes the most important loss of energy in the steam-turbine. This loss of energy is due, on the one part, to the incomplete utilization of the speed of the steam, and, on the other part, to the frictions and eddies of the stream of steam in its passage across the moving bucketing.

A second loss, the importance of which is not negligible, is caused by the frictions of the wheel and of the bucketing on the steam in the case of one or several wheels. These frictions are equally important for drum turbines where the moving bands of buckets, and the cylindrical surface of the drum rub, each in its turn, upon the steam.

In the case of a moving wheel, the work lost by its friction is proportional to the density of the fluid, to the cube of the angular speed, and to the fifth power of the diameter. As the

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density of saturated steam is sensibly proportional to the absolute pressure, then for equal diameter, the frictions are considerable for the first wheels which receive the steam at high pressure, and nearly negligible for the turbines that revolve in a medium having a pressure but little above the pressure in the condenser.

Thus, the turbine-wheel receiving steam of the pressure 147.23 pounds per square inch absolute, has a loss of work by friction a hundred times greater than the same wheel revolving in steam of the condenser-pressure of 1.55 pounds per square inch absolute.

The first wheel of a turbine must consequently have a diameter very much less than its last wheel; the different diameters being calculated in such a manner as to sensibly equalize the loss by friction.

An analysis of this friction of the wheels, reveals the cause of the excellent economy of low pressure turbines utilizing exhaust steam at atmospheric pressure according to the ingenious system invented by Mr. Rateau, while, with piston steam-engines, it is the low pressure cylinder that gives the poorest economy, either by the increase of its frictions or by its thermic losses. On the contrary, in the steam-turbine, it is the low pressure wheels that give the maximum economy.

The importance of the friction of the wheels, has often been objected to in the multicellular system. The experiments made for determining the law of the frictions, and the results obtained with machines calculated according to the coefficients furnished by practice, have justified the indications of theory that machines of this nature can be constructed in which the frictions will be represented by only from 2 to 4 per centum of the total work. These figures are comparable with those that have been obtained with drum turbines, in which account must not only be taken of the frictions of the moving buckets, but also of the frictions of the cylindrical surface of the drums.

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LOSSES BY EDDIES.

The losses by eddies can be considerable, unless precautions are taken to limit them.

In the reaction turbines, as has previously been stated, the fluid which gradually expands in the moving buckets has naturally a tendency to return backward to the orifices by which it entered these buckets. To prevent this effect, these buckets must be kept constantly filled with gaseous steam flowing in the same direction that they move. Now this result can only be obtained by feeding the buckets in a continuous manner. Consequently, the reaction turbine requires, as has previously been said, feeding around the entire periphery of its wheels.

Experiment has shown that if partial injection of steam be applied to a reaction turbine, the economy will be very considerably lowered. Partial injection is possible, then, only for turbines of action.

In the turbines of action with one wheel, where the total fall of pressure is utilized by a single distributor, the form of the tuyere is necessarily convergent-divergent; unfortunately, the divergent part is always an organ of bad economy. Phenomena of the adhesion of the steam to the inner surface of the tuyere, causes losses of economy which the author has often measured, and which attains in the best conditions from 10 to 12 per centum of the total energy contained in the steam. This constitutes a grave inferiority for turbines of one wheel, whatever may be their mode of construction, unless their employment be restricted to minimum falls of pressure, the absolute high pressure not exceeding double the absolute low pressure.

Another cause of eddies exists in the trajectory of the steam leaving the moving bucketing.

If the turbine be so accurately constructed that the steam escaping from it exactly meets the openings in the following distributor, the principal part of the speed remaining in the steam current will be utilized. This remaining energy is not

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an important part of the total energy, but in multicellular turbines it reaches 5 or 6 per centum.

On the contrary, if the steam current leaving the turbine strike an opposing surface, its remaining speed will be entirely lost, and the corresponding energy will be wasted in concussions and eddies.

In multicellular turbines using the pressure in successive steps or stages, it is indispensable to so exactly fit the successive distributors that the remaining speeds of the steam will be utilized.

CALCULATION OF THE ECONOMY.

For a given wheel and a given fall of pressure, that is to say, for a speed of efflux of the steam determined in advance, the hydraulic economy varies with the peripheral speed of the wheel. A curve can then be traced of the hydraulic economy in function of the peripheral speed, which permits the ascertainment in each case, of the value that can possibly be attained in practice.

After the hydraulic economy has been determined, there must be deducted from it the losses by leakage, by friction, and by eddies.

Take, for example, a Rateau multicellular turbine furnishing 1,500 effective horsepowers to the shaft at the speed of 1,500 revolutions per minute.

By the aid of the practical coefficients obtained from experiment, the hydraulic economy of such a machine can easily be calculated; it is equal to 0.69. The losses by steam leakage and by the frictions of the pillow-blocks, absorbs 1½ per centum of the power. The losses due to the frictions of the wheels upon the steam are 2½ per centum, making a total of 4 per centum.

The net economy on the shaft at the speed indicated, will then be $0.69 \times 0.96 = 0.662$.

The following table shows the comparative elements of the economy of the same turbine at speeds of 1,500 and of 2,000 revolutions per minute.

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CALCULATIONS OF THE ECONOMY OF A RATEAU MULTI-CELLULAR TURBINE OF 1,500 HORSEPOWERS.

—	At 1,500 revolutions per minute.	At 2,000 revolutions per minute.
Hydraulic economy, fraction	0.69	0.76
Losses by steam leakage and by the friction of the pillow-blocks, in per centum	1.5 } 4.0	1.8 } 7.7
Losses by the friction of the wheels on the steam in per centum	2.5	5.9
Net economy on shaft, fraction	$0.69 \times 0.96 = 0.662$	$0.76 \times 0.923 = 0.702$
Steam pressure absolute, in lbs. per square inch	206.12	
Back pressure absolute, in lbs. per square inch	1.16	
Superheating, in Fahr. degrees	212	
Consumption of steam in pounds weight per hour per theoretical horsepower	6.83	
Consumption of steam in pounds weight per hour per effective horsepower	$\frac{6.83}{0.662} = 10.32$ lbs.	$\frac{6.83}{0.76} = 9.74$ lbs.

From the above table will be perceived how greatly the economy increases with the speed.

There is given in the above table the calculation of the consumption of steam per hour per effective horsepower, at the two speeds stated.

This consumption, the steam being superheated 212 Fahrenheit degrees, is 10.32 pounds at the speed of 1,500 revolutions of the turbine per minute, which corresponds to about 9.04 pounds of steam per hour per indicated horsepower.

At 2,000 revolutions per minute, the consumption of steam per hour per indicated horsepower will be about 8.60 pounds.

From these figures there appears that the practical results which can be obtained from a multicellular turbine of 1,500 horsepowers, are comparable with those given by the best piston steam-engines that actually exist.

The calculation stated of the economy, it must be well understood, is according to practical coefficients which only practice can give. But in the case of the piston steam-engine, the cylinder condensation is uncertain and often erroneous,

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while, in steam-turbines, the continuity of the flow of the steam permits the establishment of a theory of great exactness, applicable to all cases of practice, once the fundamental coefficients are determined.

ECONOMIES OBTAINED WITH DIFFERENT SYSTEMS OF STEAM-TURBINES HAVING MULTIPLE WHEELS.

The analyses that have just been made of the various elements which influence the economy of steam-turbines, enables the calculation to be made of the economies obtainable with different systems of them.

In order to illustrate the preceding considerations by a comparative example, they will now be applied to the two principal systems of reaction turbines having multiple wheels; the turbine with successive falls of pressure, and the turbine with successive falls of speed.

The Rateau multicellar turbine is essentially a turbine with the pressure in successive stages or successive falls of pressure; the speed with which the steam circulates is constant in the different cellules, and as the economy dffers but little from one wheel to another, the work done per wheel remains constant from one extremity of the turbine to the other extremity.

The turbine with the Curtis groups of wheels, is, on the contrary, like the Terry turbine, a turbine with successive falls of speed. The pressure is constant in each group of wheels, but the speed of circulation of the steam rapidly decreases from one wheel to the other.

Further, if the principle of successive speeds in stages be rigorously applied, such a turbine would act only, as Mr. Mortier has shown, like a single group of wheels for taking up the entire speed of the steam.

As has been previously stated, the Curtis turbine had, at first, 3 wheels in each group, which number has been reduced to 2. The comparison of the Curtis turbine with the multicellular turbine, has been made for the former with two groups of wheels.

In making the comparison of the different turbines, there

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is first compared the purely theoretical wheel consisting of a disc furnished with buckets recurved in half-arcs of a circle receiving the steam and permitting it to escape in directions perpendicular to the axis, and at 180 degrees from each other. The turbine with speeds in successive stages, comprises a group of two similar wheels; and the multicellular turbine has only a single wheel.

In both cases, the speed of efflux of the steam, which is to be utilized, will be the same, and, in order to simplify the calculations, will be assumed as equal to unity. The ratio of the peripheral speed of the wheel to the absolute speed of the steam being designated by the letter E, this value will then be that of the peripheral speed. The steam impinges on the wheel with the speed unity; it enters the first bucket with a relative speed W_0 that slackens little by little from its entering to its leaving the canal or passage of the buckets, the coefficient of this reduction being less than unity. The relative speed when leaving is then

$$W_1 = \lambda W_0.$$

From these data, and with the aid of Mr. Rateau's theory previously cited, and which experiment has confirmed, the calculation can easily be made of the total push or pressure on the wheel according to the theorem of the quantities of movement; the work done by the steam is thus immediately deduced, as well as the value of the economy, by the ratio of this work done to the disposable energy in the current of steam at its entrance into the first wheel.

The relative speed of the steam on leaving the first wheel, allows the calculation to be made of its absolute speed at that leaving. This absolute speed becomes reduced during the passage of the steam across the second distributor, and the value of the speed of the steam is thus obtained at its entrance to the second wheel.

The same method enables the push, the work done, and the economy, of the second wheel to be known.

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From which can be obtained the aggregate economy of the two wheels.

From these values the maximum economy can easily be calculated by varying the peripheral speed of the wheel, or its ratio to the absolute speed of the steam. The result of the calculation is given in the following table for three values of the coefficient of the reduction of speed of the bucketing: namely, 0.70, 0.75, 0.80.

Absolute speed of the steam $V_0 = 1$.

Ratio of the peripheral speed U of the wheel to the speed of the steam $E = \frac{U}{V_0}$.

——	Turbine of action with groups of two wheels.			Multicellular turbine.		
Coefficient of reduction of the relative speed in the bucketing λ	0.70	0.75	0.80	0.70	0.75	0.80
Value of E corresponding to the maximum of work.....	0.233	0.235	0.239	0.50	0.50	0.50
Maximum value of the hydraulic economy.....	0.590	0.642	0.700	0.850	0.875	0.900

The maximum economy that can be furnished by groups of two wheels, is seen to be very inferior to the economy which is given by the multicellular turbine wheel functioning equally in the conditions of the maximum.

This inferiority can be presented in a still more striking manner.

Suppose that the groups of two wheels with successive stages of speed, are transformed into two wheels with successive stages of pressure by separation by means of a diaphragm.

Starting from the figures obtained in practice, the ratio of the peripheral speed of the wheel to the absolute speed of the steam, is 20 per centum. For the same absolute speed of entering the wheel, the fall of pressure being divided in such manner as to give equal work on the two wheels, it is certain that the preceding ratio in the second system rises to more than 28 per centum.

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Accordingly, it is easy to calculate that the hydraulic economy of the group with speeds in steps or stages will be 0.688 for the most favorable value of the coefficient $\lambda = 0.80$, while the same group with successively divided pressure will have an economy of 0.767.

In taking for the coefficient of the slowing λ the value 0.70, the hydraulic economy will be 0.580 for the system of successively divided speeds, and 0.704 for the system of successively divided pressures.

Evidently, then, and without taking into the account the faculty which the system with successively divided pressure has of a great deal increasing the number of wheels, which again increases the ratio of the speeds E, and, consequently, the hydraulic economy, the simple introduction of the multicellular principle allows the economy of a turbine with successive stages of speed to be considerably increased.

The preceding reasoning supposes that the bucketing of the turbines is arranged in demi-circumference. In practice this cannot be the case, and, in order to calculate the hydraulic economy it is important to take into the account the angular inclination of the buckets and of the current of steam.

Let there be considered two turbines, one with successively divided speeds, and in groups of two wheels; and the other the multicellular system. The first with a ratio of the peripheral speed to the speed of the steam of 20 per centum, which is the practical case; and the second with a ratio of 35 per centum, which is the case of the Rateau turbine.

Suppose the buckets to have the same inclination in the two turbines, namely, 30 degrees to the plane of the wheel, and that the angle of the steam current or jet is 20 degrees when leaving the distributor.

In taking the total work contained in the steam for unity, there is found that for a value of the coefficient of 0.70, the useful work in the preceding wheel is 0.494, and the useful work in the second wheel is 6 per centum, or the total of 0.554 for the two wheels. In the Rateau turbine, on the contrary, the useful work in the first wheel is 0.700, and that due to the remaining speed of 6 per centum, making a total of 0.760.

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In these conditions, the Rateau turbine gives a supplement of 37 per centum of work.

If, on the contrary, the supposition be made that the coefficient λ is more favorable and attains 0.80, a value never yet obtained in practice, there will be found that the work utilized in the two wheels with speed successively divided is 0.646, and in the wheels with pressure successively divided 0.783, or a supplement of still 21 per centum.

This last discussion, like the preceding one, leaves no doubt on the primordial fact that the hydraulic economy of a turbine with successive falls of pressure, multicellular, is always superior to that of a turbine with successive decreases of speed in the steam.

COMPARATIVE ECONOMY IN FUNCTION OF LOAD.

In Figure 15 are reunited the comparative curves of the total economy in function of the load, and at constant speed, of a certain number of groups of generators of electricity by means of different systems of steam-turbines.

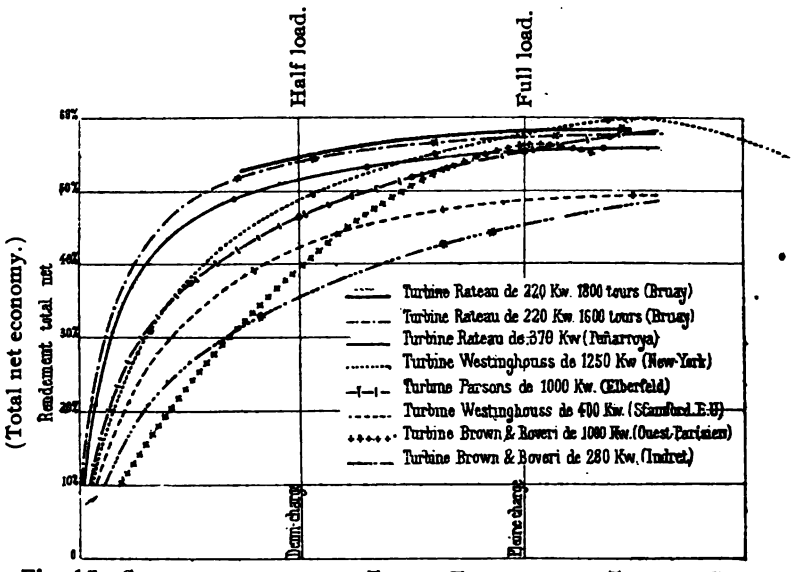

Fig. 15—COMPARISON OF THE TOTAL ECONOMY OF TURBINE-DRIVEN ELECTRICAL GROUPS, IN FUNCTIONS OF THE LOAD.

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These curves have been traced by calculating the economy according to the experimental results that have been published. The abacus has been used for that purpose.

In a general way, an examination of these curves makes known that the total economy increases in function of the load until a certain maximum is attained; beyond which point it decreases.

They include only two groups of Rateau turbines: one of 220 kilowatts at low pressure, and the other of 400 kilowatts at normal pressure.

The other curves belong to different turbines of the Parsons system, constructed either by the inventor or by the firms Brown-Boverie, and Westinghouse.

The economy of the Rateau groups is generally superior, for equal power, to that of the other machines, and with feeble loads, the curve is maintained in the upper regions.

It is scarcely necessary to say, that these curves do not exactly give the real economy of the turbine separate from its generator of electricity, because they include, as is well understood, the economy of that generator.

Nevertheless, from the practical point of view, the important fact is the net economy of the turbine and of the dynamo or alternator when combined.

An other remark that results from these curves is that the steam-turbines have generally a satisfactory economy with feeble loads.

When the steam is throttled and its pressure consequently lowered, the hydraulic economy of some of the wheels of the turbine increases, and therefore recuperates a more important proportion of the work, although the disposable energy in the steam was diminished.

On the other hand, the frictions of the wheels on the steam diminish with diminution of the mean density of the steam.

These phenomena do not happen with the piston steam-engines, and, consequently, their economies with feeble loads are inferior to those of steam-turbines.

Figure 16 gives comparative curves of the maximum econ-

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omy at full load for different systems of turbines. The abscissas represent the value of the full load up to 4,000 kilowatts; the ordinates represent the total economy at full load.

The appearance of these curves is very different for the different systems. There must not, however, be drawn too precise consequences from the calculations made by means of

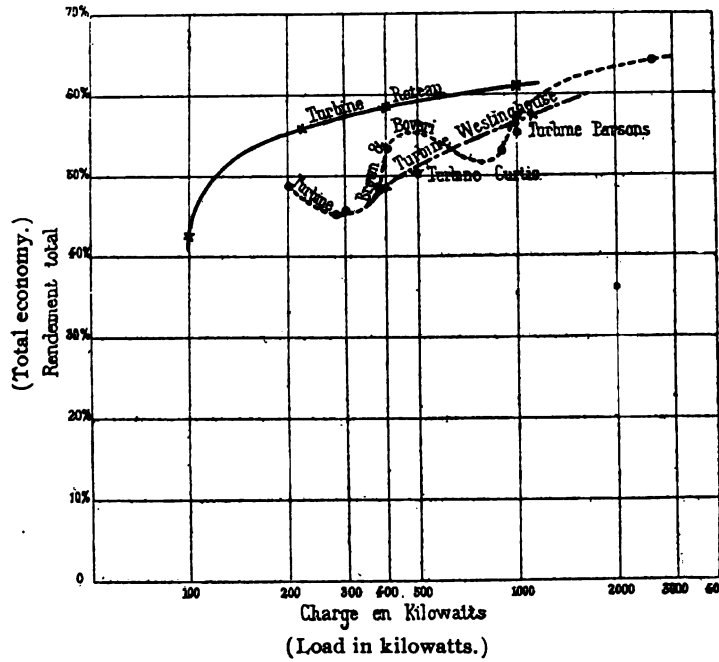

Fig. 16.—MAXIMUM ECONOMY OF ELECTRICAL GROUPS WITH DIFFERENT SYSTEMS OF TURBINES.

the too scanty information published for the different machines, and which should be completed whenever the data furnished by their constructors shall be more numerous.

APPLICATIONS OF THE RATEAU TURBINE.

I.—GROUPS FOR THE GENERATION OF ELECTRICITY.

The most important application made of the Rateau turbines up to the present time, has been for the operation of generatrices of electricity either with continuous current or with triphased current.

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CONTINUOUS CURRENT.

Dynamos of continuous current operated directly by steam-turbines, have quite special difficulties of construction. Their great angular velocity compels their establishment with a small number of poles; whence results that the reaction of the introduction is considerable, and that the employment of special systems of compensation is necessary to prevent the disastrous effects of a bad commutation of the brushes.

With the Parsons turbine, the firm of Brown has principally employed a system of compensation formed by supplementary spirals placed in the polar pieces and fed or supplied in series.

The workshops of Sautter, Harlé and Co. have had recourse to a system of complementary inducting poles, fed in series, and creating a magnetic field in the plane in which the commutation is made.

This first difficulty being overcome, the dynamos offer another, which is the construction of the collectors. This delicate part of the machine should be made with the minutest care, because it is composed of heterogeneous pieces of metal and of isolants which are very difficult to center and to adjust rigorously for preventing the effects due to the centrifugal force.

The centrifugal force is, in fact, considerable in generatrices operated by steam-turbines; it attains, and often surpasses, at the periphery, 2,000 times the weight.

Also, the equilibrating of all the revolving parts should be minutely cared for. Special arrangements for the statical and for the dynamical equilibrating have been foreseen.

With dynamos of continuous current, the vibrations can render the functionment quite impossible by reason of the sparks they produce in the brushes if the latter momentarily quit the collector.

The peripheral speed of the collectors being considerable, the effect of the sparks is a great deal more injurious than in the ordinary dynamos, and it becomes indispensable to nearly completely avoid fire at the brushes.

Generally, the brushes are metallic, but in certain special

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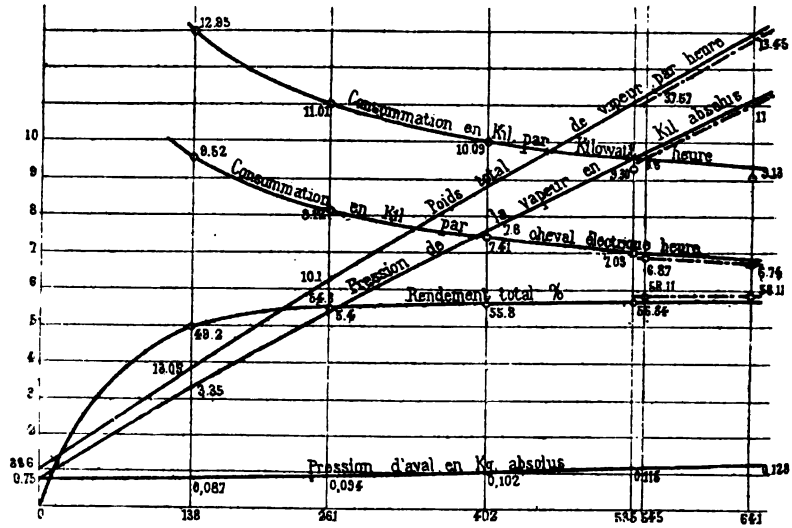
cases, notably with turbines fed by low pressure turbines where the angular velocity is much less, it is possible to employ brushes of carbon.

Always, as well with brushes of carbon as with metallic brushes, perfect equalibration of the moving parts is indispensable.

The designs of the construction of the new generators of electricity operated by Rateau turbines, have been made under the direction of Mr. Sacquet, Engineer of the Electrical Service in the establishment of Sautter, Harlé and Co.

Figure 18, Plate V, represents a group of 500 electrical horsepowers with continuous current: the turbine made in two bodies operates directly two dynamos placed end to end and arranged for three wire distribution.

Several of these machines have been made for the metallurgical works of the Society of Pennaroys, in Spain. Their rotary speed is 2,000 revolutions per minute.



(Power in electrical horsepowers measured at terminals.)

Fig. 17.—TURBINE DYNAMOS OF 590 HORSEPOWERS. CURVES OF PRESSURE, CONSUMPTION AND ECONOMY, AT A CONSTANT SPEED OF 2,400 REVOLUTIONS, IN FUNCTIONS OF THE POWER.*

* The curves ——— . —— . ——— correspond to the operation with steam superheated about 10° cent.

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The curves of consumption of steam, and of economy, are shown in Figure 17 which gives the results of the trials. It will be observed that at the speed of 2,400 revolutions per minute, the group was able to furnish up to 640 electrical horsepowers which represent about 700 effective horsepowers on the shaft. The total economy rises to 58 per centum with the steam slightly superheated.

The curves also show the remarkable proportionality of the total weight of steam consumed to the useful load; this weight of steam being sensibly represented by a straight line in function of the load.

With vacuum, when the machine does not do any external work, the dynamos being excited, the consumption of steam represents 9 per centum of its consumption at full load, a remarkably low figure showing how feeble are the frictions in the Rateau turbine.

An analagous group of 500 electrical horsepowers, operated by a piston steam-engine, consumed with vacuum 20 to 22 per centum of the weight of steam consumed at full load.

The consumption trials which have enabled the curves in question to be drawn, were made by Mr. Wissler, Engineer of the technical investigation department of the previously mentioned workshops, under the direction of which the plans were made for the execution of the different Rateau turbines manufactured in them.

There can also be remarked concerning these curves, the proportionality of the absolute pressure of the steam when entering the turbine to the useful load.

The turbine can be said to be a true counter of the consumption of steam, and in order to know the consumption at any given load, all that is required is to have measured the consumption for two other loads, the curve being sensibly a straight line.

The following table shows the comparative results of calcutions made by Mr. Rateau for a group of 500 horsepowers, and the results of trials with full load. It shows, also, what would be the consumption by the same turbine per electrical horse-

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power, functioning with a higher pressure of steam and a better vacuum.

It also shows for this group the increased economy for higher powers.

A RATEAU TURBO-DYNAMO OF 500 ELECTRICAL HORSEPOWERS.

——	Results of the calculations made by Mr. Rateau.	Results of the experiments made.	Results under the most favorable conditions.
Absolute pressure of the steam, in pounds per square inch	142.223	136.541	213.345
Absolute pressure of the exhaust, in pounds per square inch	2.844	1.636	1.159
Revolutions per minute	2,000.000	2,000.000	2,400.000
Consumption of steam, in pounds weight per hour per electrical horsepower	17.637	15.653	11.861 (Per kilowatt, 16.138)
Electrical horsepowers developed	500.000	510.000	530.000
Economy, total net	0.550	0.563	0.581

A general section of the 500 horsepowers turbine in two bodies, is represented in Figure 18. It shows the entrance of the steam, and the exhaust, for the two bodies; also, the arrangement of the pillow-blocks, all of them being outside

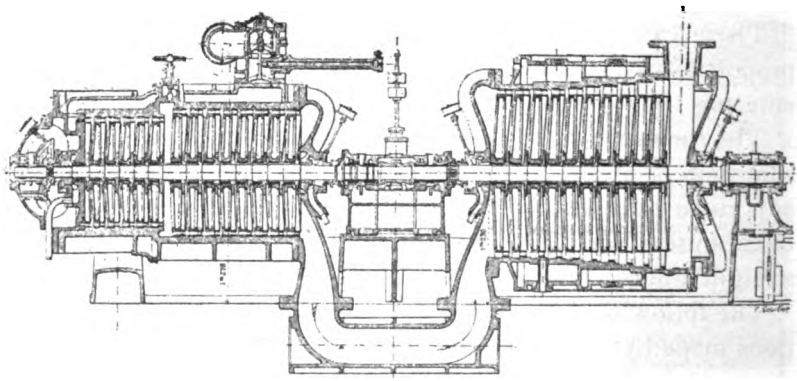

Fig. 18.—SECTION THROUGH A RATEAU TURBINE OF 500 HORSEPOWERS.

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of the casings or shells. The lubrication is made in the ordinary way, the forcing of the oil upon the journals, as originally done by special pumps, not being found necessary.

The division of the turbine into two bodies acting upon the same shaft is not necessary: on the contrary, the two bodies can be placed on two parallel shafts, one on each shaft, the two bodies being connected by an intermediate pipe conveying the steam from one to the other.

This arrangement is represented in Figure 19, Plate V, for an ensemble of two connected 500 horsepowers bodies upon two parallel shafts functioning in a steel works in Russian Poland.

The two dynamos are combined in quantity, while the two bodies are in series.

The functionment of this type of turbine is as satisfactory as that of the equivalent turbine in a single body. The first cost is a little greater, but it has the practical advantage of allowing the separate functionment of one of the bodies with its dynamo, in the case of an accident to the other.

In regard to space occupied, according to the case, there is an advantage in employing the group with a single shaft, or the group with two shafts.

The utilization of two dynamos permits, especially as regards the distribution of the force in a workshop, the employment of the three wire system which enables a sensible economy to be realized in the cost of the canalizations.

The working of dynamos by turbines is particularly well adapted to the distribution with three wires.

TRIPHASED CURRENT.

The construction of generatrices with triphased current in relation to turbine, presents less difficulty than dynamos of continuous current. The suppression of the collector is an important advantage as it permits in practice the possibility of higher angular speeds for the alternators than for the dynamos of continuous current.

Sautter, Harlé and Co. have tried successively three types of

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alternators: the turning iron alternator; the turning inducing alternator; and the turning inductor alternator.

The turning iron alternator would be from the mechanical point of view, the ideal generatrice for the steam-turbine, did not its very construction restrict the conductor to only two polar portuberances equivalent to four alternate poles. Now when it operates a machine giving 50 periods per second, evidently the speed of rotation corresponds to 1,500 revolutions per minute; but, for turbines of less than 1,000 horsepowers, the speed of 1,500 revolutions per minute is too small and leads to a turbine of disproportionate dimensions and of mediocre economy.

But this is not the only disadvantage of the iron turning system: with equal weight, the magnetic leakage and the fall of tension in these machines are more considerable than for the other types of alternators.

With equal quality, the weight of these machines is notably greater.

These disadvantages are very regretable, because the turning iron alternator, allows the construction of an entirely massive moving part that can easily be perfectly equilibrated at the moment of its construction, and which has no chance of being injured.

The second type of alternator, is the turning inducing alternator. It is according to this type that Parsons has constructed the greatest number of electrogen groups functioning in England.

The turbo-alternator of the town of Elberfeld, well known by the complete experiments made with it, is properly of the same type.

With a turning inducer, a machine of minimum weight can be constructed of which the inducing reaction is the feeblest, being given as the inductors can be saturated.

But there is a difficulty; that of constructing the inducers, which have to turn for high tensions at very considerable peripheral speeds; the place lacking for the isolants, and the

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effects of accumulation due to the centrifugal force, can, after a while, compromise the isolation.

The turning inductor alternator comprises an inducer fixed in sheet iron, and a polar turning inductor, constructed in such a manner as to be able to resist great centrifugal forces.

The tension of the current in the inductor being feeble, since it acts only as the current of excitation, a good isolation can be more easily obtained.

The disadvantage of this system consists in the small number of its poles. For a frequence of 50 periods, if it be desired to make 3,000 revolutions per minute, only two poles are necessary; at 1,500 revolutions per minute, four poles are sufficient.

These numbers of poles, for a frequence of 40 periods, correspond to 2,400 and to 1,200 revolutions per minute.

Now the alternators having a small number of poles, have a strong inducing reaction, notably for the bipolar alternators, the induction cannot be saturated, because the magnetic losses would become too large.

The fall of tension in the alternators of two or four poles is then considerable when passing from working at a vacuum to working with a full load and constant excitation. It becomes necessary, when the variations of the load are frequent on the iron net-work that feeds the machine, to provide an automatic system of compensation of the voltage, and above all for the bipolar alternators.

Figure 19 represents a group of 1,000 electrical horsepowers, 700 kilowatts at 1,000 volts and 1,500 revolutions per minute. The alternator is of iron and turning, and the turbine is of the plan first designed, consequently with interior pillow-blocks.

This machine which was designed and constructed in 1900, has given excellent results from the mechanical point of view, but the defects of the iron turning alternators are very sensible: great fall of tension; great magnetic leakage.

The turbine is of the same design as the one made for the Torpedo Boat *243*, of which more will be said later on.

The curves given in Figure 20 show the economy of that

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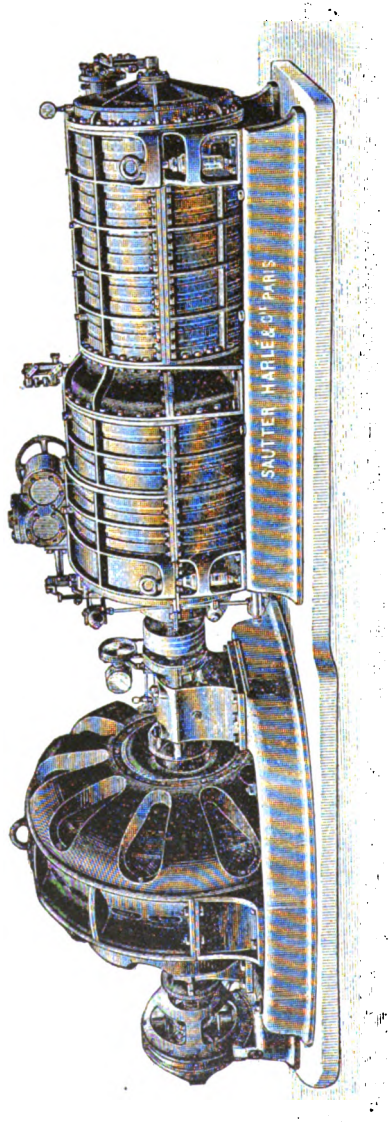

Fig. 19.—Turbo-Alternator of 1,000 Horsepowers.

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design of turbine at the speed of 1,500 revolutions per minute. This economy has been measured according to the consumption of steam and according to the measure of the economy of the alternator itself, calculated according to a method devised by Engineer Blondel.

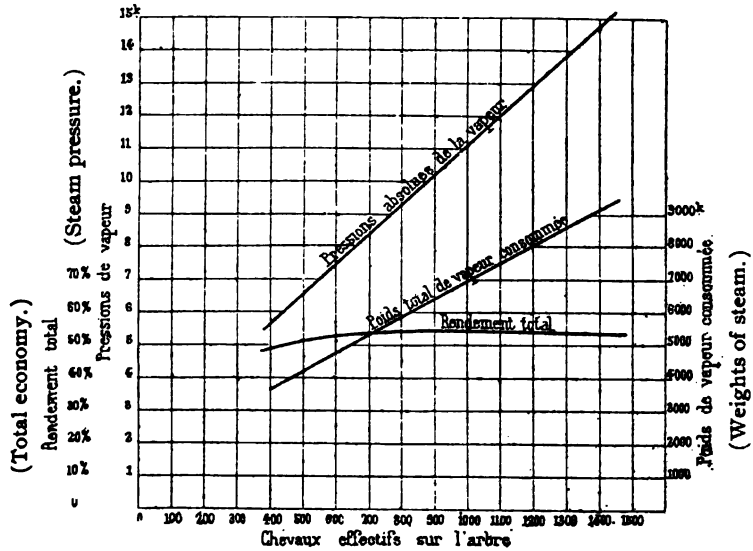

Fig. 20.—RATEAU TURBINE OF 1,000 HORSEPOWERS. TOTAL ECONOMY AND STEAM CONSUMPTION.

The design of the turbine in question, was made for a speed of 1,800 revolutions per minute: the curves of economy relate to the speed of 1,700 revolutions per minute.

This type was too much reduced as regards dimensions, because of the application that was desired to be made of it in marine propulsion; but, nevertheless, it is interesting to find that the result obtained with full load during the experiments in 1903, correspond to a value of 0.535 as the net economy of the turbine, while the calculations made by Mr. Rateau in 1898 assigned an economy of the value of 0.530.

Figure 21 represents a group of 400 electrical horsepowers with 5,500 volts at the speed of 3,000 revolutions per minute.

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This machine is now functioning in a station of "The Electrical Company of the Loire."

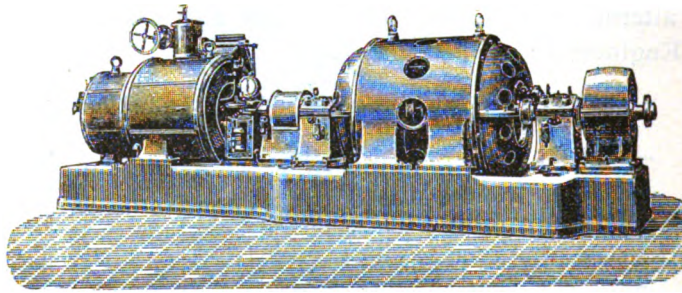

Fig. 21.—TURBO-ALTERNATOR OF 400 ELECTRICAL HORSEPOWERS, 5,500 VOLTS AND 3,000 REVOLUTIONS.

The following table shows the results of the calculations made by the builders, and the results obtained from experiment.

TURBO-ALTERNATOR OF 400 ELECTRICAL HORSEPOWERS.—RATEAU SYSTEM.

——	Calculated by the builders.	Results of the experiments.
Pressure of the steam, in pounds per square inch above zero	170.678	168.403
Pressure of the exhaust, in pounds per square inch above zero	2.133	2.845
Revolutions per minute	3,000.000	3,000.000
Electrical horsepowers developed	370.000	388.000
Economy	0.470	0.487
Consumption of steam, in pounds weight per hour per electrical horsepower	18.739	19.224
Consumption of steam, in pounds weight per hour per kilowatt	25.573	26.014

The economy obtained would have been increased by augmenting a little the dimensions of the turbine which contains only a dozen working wheels; but is still superior to that of any group of turbines of the same power of any other system.

It is interesting to observe that this machine debits upon a wire net work whose factor of power is very feeble, not surpassing as a mean from 0.55 to 0.60. This net work feeds

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small synchronous motors employed in the ribbon factories at Saint-Étienne.

The alternator being bipolar, and the factor of power being very feeble, the fall of tension with constant excitation between the vacuum and the full load is considerable; in fact, it exceeds 40 per centum. Under these conditions, it is indispensable to compensate this fall by a compounding of the alternator.

This compounding has been realized by means of a dispositive designed by Engineer Blondel which consists of the employment of an excitatrice of reaction and double excitation combined in series with a transformator. The results thus obtained are very satisfactory, and the compounding has been shown to be made with a variation of only some hundredths on the measured voltage.

At the present date Sautter, Harlé and Co. have under construction groups with triphased current of 300 electrical horsepowers at 1,000 volts; of 360 electrical horsepowers at 2,000 volts; of 700 electrical horsepowers at 5,000 volts; constructed with induced turning alternators or with turning inductor.

CONSTRUCTION OF RATEAU TURBINES IN THE WORKSHOPS OF OERLIKON.

The construction of electrogen groups, with turbines of the Rateau system, has likewise been undertaken in Switzerland by the workshops of Oerlikon.

Several groups with generators having a continuous current or a triphased current have been constructed in these well known shops.

A great deal of talk has been made of certain difficulties encountered at the commencement by the Oerlikon shops during the first constructions.

These difficulties arise, on the one part, during the construction of new electrical machines which require, as has previously been stated, the solution of problems not found in the construction of slower machines.

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On the other part, particular circumstances, such as the arrangement for the entrance of the steam into the turbine, and those of the piping, have been in the Rheinfelder workshop, the cause of incidents that have been much exaggerated by persons of opposing interest.

At the present time, the Oerlikon workshops have overcome these first difficulties, and have perfected the manufacture of steam-turbines of the Rateau system. There can be cited, notably, an electrogen with continuous current, having a power of 100 kilowatts with 3,000 revolutions per minute, installed at Ruti in Switzerland, and which has been functioning normally during 18 months.

It is important to recall that the Rateau turbine with two bodies in cascade (Figure 16, Plate IV), having 2,000 horsepowers, which has given excellent results during recent experiments, in the propulsion of a torpedo boat constructed by Yarrow in England, came from the Oerlikon workshops.

Later on, remarks will be made concerning the application of the turbine to marine uses.

II.—TURBO-PUMPS.

PRINCIPLE OF THESE APPARATUS.

The working of a centrifugal pump by a steam-turbine, seems, at the first view, to present insurmountable obstacles.

As has been shown by Mr. Rateau, in his memoir published in the "Bulletin de la Société de l'Industrie Minerale" in 1902, it is always possible to operate a centrifugal pump at considerable angular velocity by a steam-turbine, but, unless certain precautions are taken, the economy of such a combination, will be very inferior.

Beyond a certain angular speed, the blade of a centrifugal pump is not capable of delivering, the receiving itself being prevented.

This phenomenon, which is comparable to that of cavitation in the case of the blades of a propelling screw, is caused by the discontinuity in the stream of liquid which can no longer follow up the moving blades of the centrifugal pump. Beyond

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a certain debit for a given speed and a given delivery, the economy of the pump continues to decrease and finally becomes altogether defective.

The phenomenon of cavitation obliges the loading of the wheel of the centrifugal pump whenever its angular speed surpasses a certain limit for a given debit and a given height of elevation of liquid.

When the intention is to directly work the centrifugal pump by a steam-turbine, it is indispensable to load this pump by means of an auxiliary pump which produces the normal debit with a great deal less height of elevation, attaining at the most 5 to 10 per centum of the total height to be produced.

When turbines of a single wheel are employed like those of Laval, or like those which Rateau first constructed, it becomes impossible, owing to their great angular speed, to load the principal pump by placing the auxiliary pump upon the same shaft. The phenomenon of cavitation, suppressed for the principal pump is reproduced for the auxiliary pump.

For this reason, the first inventors discovered that it was necessary to reduce the angular speed of the auxiliary pump, and consequently to place on the shaft of the principal pump reducing gearing.

In this manner Laval's pumps, and the pumps of other analogous machines, are arranged.

Figure 20, Plate V, represents the first Turbo-pump invented by Mr. Rateau, and with which experiments were made in 1901.

The turbine, formed of a single Pelton wheel, had the speed of from 15,000 to 18,000 revolutions per minute, and operated a centrifugal pump of only 3.15 inches diameter.

This wheel was fed, under load, by means of another pump; the height to which the water was lifted was 997½ feet with a debit of 882.89 cubic feet per hour, and an economy, pump and turbine combined, of 0.280. The wheel would absorb under these conditions about 50 horsepowers at its axis.

In another experiment, there was debited as high as 1,518.57

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cubic feet per hour, at 862.87 feet of height, and the pump then absorbed 70 horsepowers at its shaft.

These remarkable experiments demonstrated at once, that a centrifugal pump could deliver, so to speak, to illimitable heights, and that the economy of such a combination was comparable to that of the similar groups of the best pumping piston steam-engines.

By total economy of a turbo-pump, is meant the relation of the useful energy in the water lifted, to the total energy in the steam for the fall utilized from the highest pressure in the turbine to the pressure in the condenser.

The experiments that have just been cited show the possibility of making turbo-pumps of excellent economy.

Mr. Rateau has conceived the idea of operating by means of multicellular steam-turbines, multicellular pumps which he has designed.

The Rateau pump is essentially composed of one or two bodies containing a series of movable wheels and of fixed diffusers, allowing the successive increase of the pressures for each wheel in order to reach the total pressure.

The employment of such a combination permits, for the same angular speed, the suppression of the phenomenon of cavitation, or, at least, a restriction of its limits.

On the other hand, by working the pump, thus combined, by a multicellular turbine, the speed, for the same conditions of employment, can be notably lowered, and the auxiliary pump can be placed on the same shaft as the principal pump.

DESCRIPTIONS OF DIFFERENT TURBO-PUMPS.

Figure 22 represents a turbo-pump delivering 6,356.81 cubic feet of water at a height of 688.99 feet, and absorbing at its shaft 230 horsepowers.

The turbine exhausts into an ejecto-condenser with a vertical axis; this condenser is in the same casing with the turbine. The condensing water is supplied by a small centrifugal pump by the main shaft itself.

The ensemble of this apparatus constitutes thus a small

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complete system for the elevation of water, with steam-motor, condenser, the lifting pump, and the auxiliary pump.

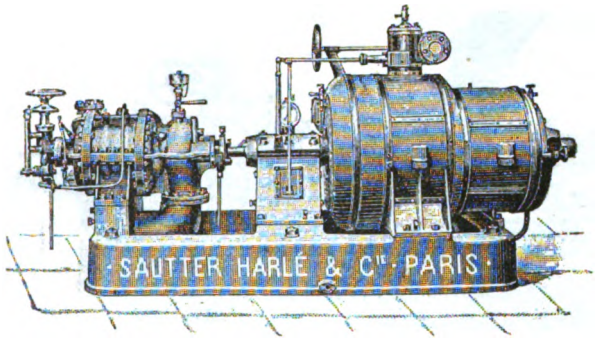

Fig. 22.—RATEAU TURBO-PUMP OF 230 HORSEPOWERS.

The following table gives the comparative results of the calculations made by the builders, and the results given by experiment.

The total net economy rises to 0.366, corresponding to a consumption of steam of 28.109 pounds per hour per useful or net horsepower at the low pressure of 90.316 pounds per square inch absolute.

RATEAU TURBO-PUMP RAISING PER HOUR 6,356.81 CUBIC FEET OF WATER 688.99 FEET.

——	Calculated by the builders.	Results of the experiments.
Cubic feet of water delivered per hour	6,356.810	6,356.810
Total height of the delivery, in feet	675.865	682.427
Revolutions per minute	3,100.000	3,250.000
Pressure of the steam, in pounds per square inch above zero	128.007	104.539
Pressure of the steam in the condenser, in pounds per square inch above zero	2.560	2.475
Total net economy	0.344	0.366
Consumption of steam, in pounds weight per hour per useful horsepower in lifting the water (condensation included)	28.219	28.109

A more powerful Rateau turbo-pump has been recently constructed for the Société des Mines de Bruay, in the Pas-de-Calais.

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This machine, represented in Figure 23, can raise per hour 8,828.900 cubic feet of water 1,181.124 feet high.

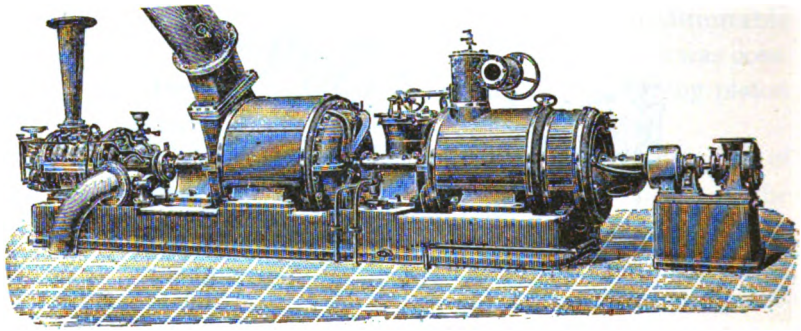

Fig. 23.—RATEAU TURBO-PUMP WITH CAPACITY OF 8,829 CUBIC FEET OF WATER AGAINST A HEAD OF 1,181 FEET.

The precisions of the calculations, and the results of the experiments are shown in the below table.

RATEAU TURBO-PUMP RAISING PER HOUR 8,828.900 CUBIC FEET OF WATER 1,181.124 FEET.

—	Calculated by the builders.	Results of the experiments.
Cubic feet of water delivered per hour..	8,828.900	9,393.950
Total height of the delivery, in feet.....	1,181.124	1,194.248
Revolutions per minute.....	2,250.000	2,200.000
Pressure of the steam, in pounds per square inch above zero.....	99.561	100.983
Pressure of the steam in the condenser, in pounds per square inch above zero.....	1.561	1.621
Total net economy.....	0.430	0.425
Consumption of steam, in pounds weight per hour per useful horsepower in lifting the water (condensation not included).....	22.046	22.707

The great economy of this pump is remarkable, as, everything included, it rises to 0.425. Further on will be seen, in relation to the consumption of steam, a total economy of over 40 per centum when steam of high pressure is used.

One of the great advantages of turbo-pumps, in addition to their excellent economy, consists in their small bulk comparably with the bulk of piston-pumps.

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There are shown in Figure 24, the comparative dimensions of the pit necessary for the turbo-pump in question, and necessary for the pit of a piston-pump of the same power.

The turbo-pump of the Bruay mines represents, in fact, 500 horsepowers on the shaft, but the dimensions of the turbine with two bodies and those of the pump itself, are so greatly below the dimensions of a corresponding piston-pump that there results an economy in the dimensions of the pits and in the volumes of the masonry nearly in the proportion of 1 to 10.

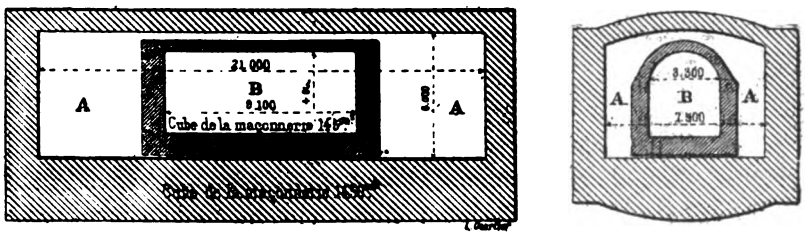

Fig. 24.—A, STEAM PISTON PUMP. B, RATEAU TURBO-PUMP.

Another turbo-pump functions in a metal-mine, and delivers per hour 3,884.716 cubic feet of water at the height of 229.663 feet.

This turbine is constructed to receive the steam at the pressure of 57 pounds per square inch above the atmosphere, but it really functions at the pressure of only 14.223 pounds per square inch above the atmosphere, the delivery being notably less than that for which the apparatus was calculated. .

With the turbo-pump very complex conditions can also be realized.

Figure 21, Plate VI, represents an apparatus constructed for a coal mine in Russia. It first raises 3,178.400 cubic feet of water per hour 295.281 feet high during the first period, then, during the second period, it raises this water 524.944 feet high.

The problem was solved by constructing a pump with three wheels, of which only two were used to elevate the water to

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the 295.281 feet height; the farther height of the 524.944 feet was effected by adding the third wheel and slightly increasing the speed of the turbine.

TURBO-PUMPS FOR FEEDING.

The turbo-pumps of the Rateau system can be advantageously applied to the feeding of boilers.

When the service is the feeding of a group of large boilers, it can be done by a turbo-pump with greater convenience and greater economy than by the auxiliary steam-pumps worked by a piston-cylinder as usually employed.

While, ordinarily, these auxiliary feed-pumps with piston-cylinders, consume from 176 to 264 pounds weight of steam per hour per useful horsepower forcing the feed-water into the boilers, a consumption that can rise to 330 pounds weight of steam per hour when the pump delivery is low, the turbo-pump for feeding does not consume more than from 44 to 55 pounds weight of steam per hour per useful horsepower.

This small consumption of steam has a greater importance than appears at first view; for at the moment the boiler is required to furnish a large quantity of steam in a given time, the quantity of steam withdrawn from it for feeding is the largest, and is then the most needed for other use.

Also, notwithstanding that its money cost is a little greater, the turbo-pump is notably the cheapest, if account be taken of the cost of the steam consumed.

Among the numerous types of steam-pumps for feeding, there can be cited a feeding turbo-pump delivering 1,412.6 cubic feet of water per hour at the height of 393.7 feet; that is to say, under the pressure of 170.7 pounds per square inch above the atmosphere, installed in the mines of Lens.

A system of automatic regulation permits this apparatus to function continuously even when its delivery is reduced to zero.

Its management by the fireman, consists in gradually opening, or gradually closing, the feed-valve.

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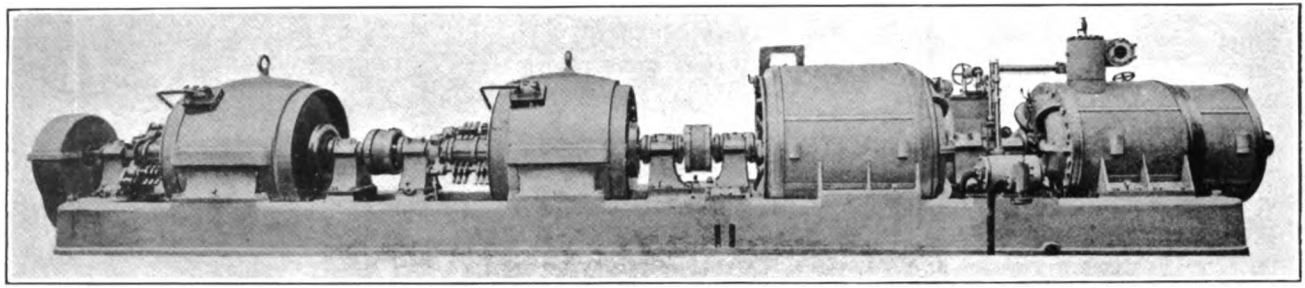

Fig. 18.—Group of 500 Horsepowers, Continuous Current.

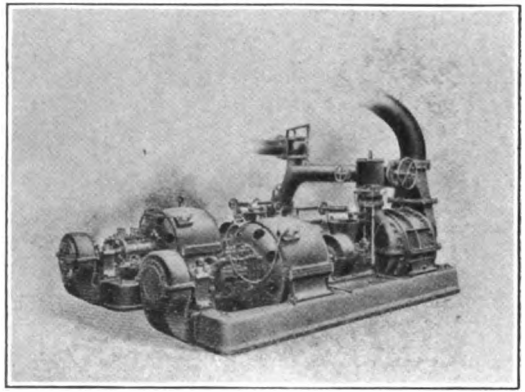

Fig. 19.—500-Horsepower Group.

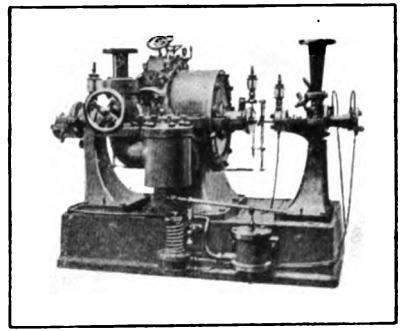

Fig. 20.—Turbo-Pump, Rateau (1901).

PLATE V.

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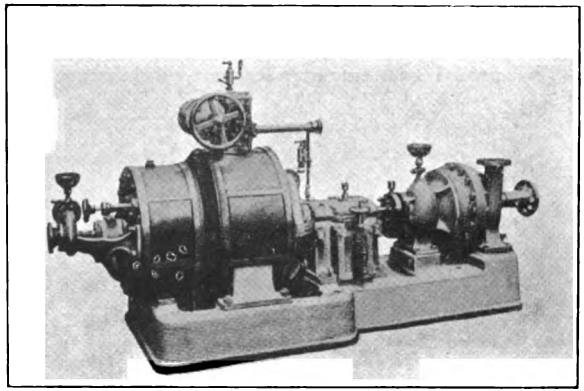

Fig. 21.—Turbo-Pump.

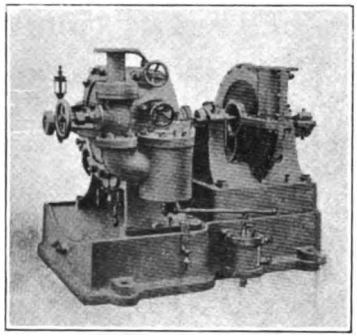

Fig. 22.—Turbo-Ventilator, Rateau (1901).

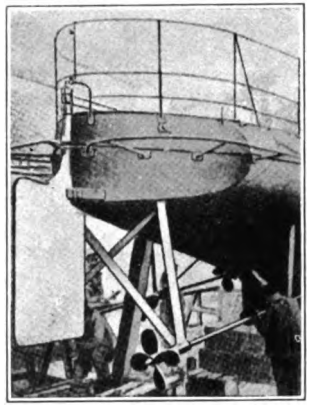

Fig. 23.—Stern of Torpedo Boat 243.

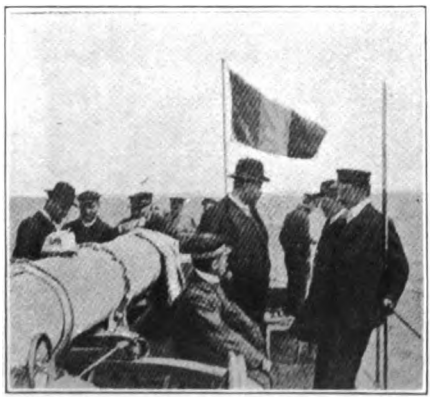

Fig 24.—Torpedo Boat 243 Under Way.

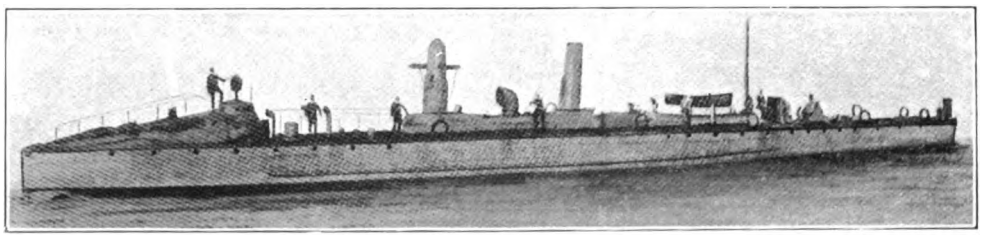

Fig. 25.—General View of Yarrow Torpedo Boat.

PLATE VI.

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TURBO-PUMP WITH LARGE DELIVERY, FOR SUPPLYING POTABLE WATER TO CITIES.

The Rateau turbo-pumps, both on account of their economy, and on account of the mechanical excellence of their functionment, are especially adapted for supplying cities with water under great pressures.

The following figures show the results that have been obtained with one of these apparatus under conditions analogous to those found in the largest cities of Europe.

Delivery of water, in cubic feet per hour	49,441.84
Total height of the delivery, in feet	459.33
Useful horsepowers applied to lifting the water	720.00
Total economy, condensation included	0.46
Pressure of the steam, in pounds per square inch above zero	213.34
Pressure of the steam in the condenser, in pounds per square inch above zero	1.14
Superheating of the steam, in Fahrenheit degrees	212.00
Consumption of steam, in pounds weight per hour per useful horsepower in lifting the water	15.09

If the above consumption be referred to the indicated horsepower, and the ratio of the useful horsepower to the indicated horsepower be taken as 88 to 100, the consumption of steam in pounds weight per hour per indicated horsepower will be 13.28.

Hence, the system in question is not inferior in any respect to the best piston-pumps actually in use.

III.—TURBO-VENTILATORS.

The great angular speeds of steam turbines render them particularly adapted for the direct working of ventilators, compressors, or blowers against high pressures.

There will be found in Mr. Rateau's memoir * already cited, the experiments made on a turbo-ventilator of his first design, represented in Figure 22, Plate VI.

This apparatus consisted of a one wheel steam-turbine of the Pelton system working directly a ventilator wheel of 12 inches diameter.

* Ventilators and centrifugal pumps for high pressures. Bulletin de la Société de l'Industrie Minérale, première livraison, 1902.

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At the speed of 20,200 revolutions per minute, this ventilator delivered 23.167 cubic feet per second under a pressure of a column of water 18.7 feet high. The useful work done in compressing the air was about 40.44 horsepowers, and the total economy of the group was 0.307, the pressure of the steam when entering the turbine being 147.9 pounds per square inch absolute.

Since that epoch, the system in question has been perfected, and a number of this apparatus has been constructed for various applications.

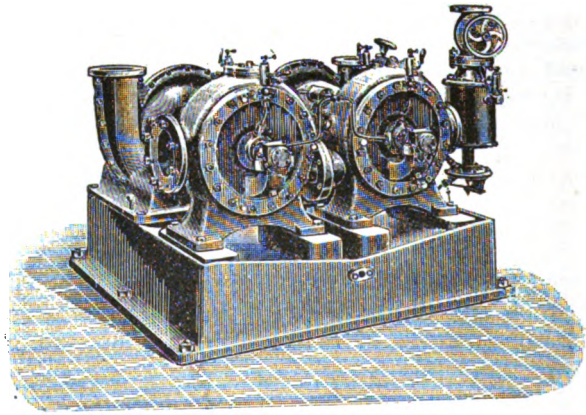

Fig. 25.—RATEAU TURBO-VENTILATOR FOR BLAST FURNACES.

One of them is employed in a sugar refinery for compressing the carbonic acid extracted from burnt lime; it is also employed in the arsenals for various purposes.

That apparatus exerts about 60 horsepowers on the shaft. An ingenious system of regulation acting upon the turbine, enables the pressure of the compression to be constantly maintained, be the delivery what it may.

The design shown in Figure 25 has been executed for the blast of a cupola in the "Forges de Chatillon et Commentry": it delivers 4,944 cubic feet per minute under the pressure of 10 inches of mercury. The compression of the air requires about 100 horsepowers.

The following table gives the results of the calculations

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made by the builders, and also the results of two series of different experiments.

RATEAU TURBO-VENTILATOR DELIVERING 70.681 CUBIC FEET OF AIR PER SECOND UNDER THE PRESSURE OF A WATER COLUMN 6.562 FEET HIGH.

	Calculated by the builders.	Results of the experiments.	
		1	2
Delivery of air per second, in cubic feet	75.929	78.401	56.858
Maximum pressure, in feet high of water column	7.546	8.858	11.713
Revolutions per minute	13,800.000	14,600.000	14,600.000
Pressure of the steam, in pounds per square inch, absolute	85.338	99.561	106.672
Pressure in the condenser, in pounds per square inch, absolute	2.845	2.845	4.381
Total net economy	0.350	0.362	0.377
Consumption of steam per hour per useful horsepower compressing air, in pounds weight	20.296	20.851	19.870

The total economy of the above apparatus rises to the remarkable figure of 0.377.

This steam-turbine is composed of two wheels of the Pelton system joined in series, each working a distinct shaft. On each shaft a ventilating wheel is placed, giving the total pressure and one half of the delivery. The orifices of the delivery are joined in quantity.

The principle of the apparatus permits the construction of machines of still greater power, replacing advantageously the usual blowing engines.

To be convinced of this latter, consider Figure 26 representing the spaces occupied by the apparatus in question and by the equivalent usual blowing engine.

Another design, now in construction, gives a delivery of 3,445 cubic feet of air per minute under the pressure of a column of water 13 feet high.

Finally, in a few months will be finished the construction of a turbo-compressor of 350 horsepowers at the shaft, compressing the air to the pressure of 85 pounds per square inch

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above the atmosphere. This apparatus is worked by a low pressure turbine utilizing the exhaust steam of a piston steam-engine, pumping water from a mine.

The turbine is in two high pressure bodies allowing it to function alone during the hours that the pumping engine is not working.

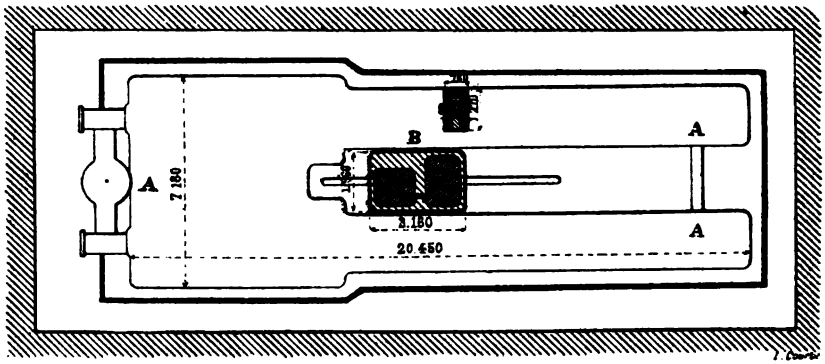

Fig. 26.—A, Ordinary Blowing Engine (1,765.3 Square Feet). B, Rateau Turbo-Ventilator (75.3 Square Feet).

This, quite a special branch of manufacture of the application of steam-turbines to the compression of air, has a great future, not only as regards mines but also for metallurgical operations, the Rateau apparatus advantageously replacing all other blowing engines of all kinds in steel works and in smelting.

LOW PRESSURE STEAM-TURBINES.

The utilization of the exhaust steam escaping from steam-engines into the atmosphere, has been the object of many designs made by Mr. Rateau.

The excellent economy of turbines using low pressure steam has enabled him to show that the utilization of this exhaust steam, impossible with piston steam-engines, is easily accomplished by means of additional steam turbines.

If steam be taken of atmospheric pressure, representing about 26.6 cubic feet per pound weight, and if this steam be expanded to the pressure in the condenser, or to about the

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pressure of 1.4 pounds per square inch absolute (see the abacus), there will be found that the weight of steam consumed per theoretical horsepower is about 17 pounds per hour.

Now, the multicellular steam-turbine in proper adjustment and utilizing the steam in these conditions, has an economy surpassing 60 per centum, and can attain even 65 per centum.

The effective horsepower upon the shaft under these conditions, represents a consumption of about 26.8 pounds weight of steam per hour.

If the turbine be replaced by a piston steam-engine, not only will the dimensions of this motor be, for equal power developed, enormously disproportionate, but its economy, by reason of its frictions and thermic losses, will be extremely low.

The dimensions of the turbine, on the contrary, are far from increasing in proportion to the volume of the steam to be utilized. Only the sections of the distributors should be calculated so as to allow the total flow of steam to pass at a convenient speed.

The steam turbine, therefore, as opposed to the piston steam-engine, is in comparatively the same conditions as the centrifugal pump is to the piston-pump.

Struck by these advantages, Mr. Rateau sought to combine the turbines with stationary steam-engines without condensation; that is to say with piston steam-engines pumping out mines, and piston steam-engines working rolling mills.

The powers which the apparatus represent are considerable, and the recuperating energy in the lost steam enables an important economy to be realized in the products of a factory.

Nevertheless, there is one difficulty: the machines in question, being of intermittent functionment, the interrupted flow of steam from them has to be regulated, as otherwise the turbine could not be employed under these conditions.

The ingenious accumulator-regenerator of steam, designed by Mr. Rateau, has completely solved this problem. It transforms a discontinuous flow of steam into a continuous flow, whatever may be the prime motor.

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Mr. Rateau has demonstrated* that not only the utilization of the exhaust steam was advantageous in the case when that steam escaped into the atmosphere, but even when it was condensed: there is an advantage in recuperating the energy of steam of atmospheric pressure down to the pressure in the condenser of a steam-turbine over effecting that recuperation in the large cylinder of a piston steam-engine.

Thus, not only is the use of the turbine rational in the cases of the pumping engines of mines, and of the engines of rolling mills, but it will be shown to be so in many steam-engines of multiple expansions.

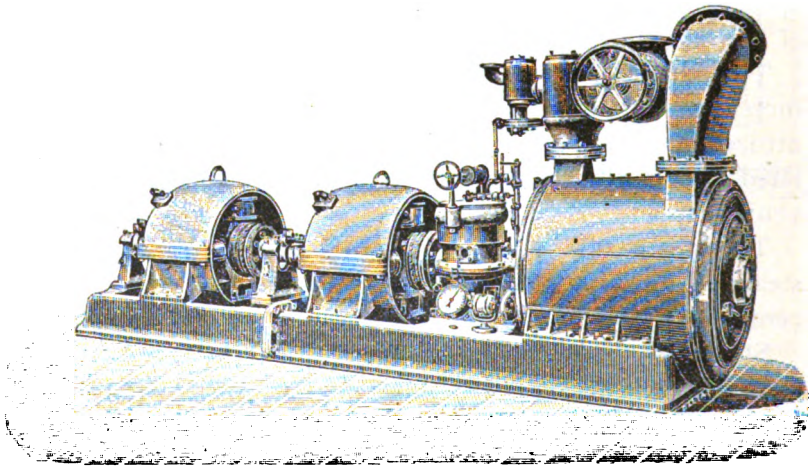

Fig. 27.—Low Pressure Turbo-Dynamo Group of 300 Horsepowers.

Figure 27 represents a low pressure turbo-dynamo group of 300 horsepowers erected in the mines of Bruay, in 1902. The turbine, of one body, uses the exhaust steam of a piston pumping engine therein, and expands it down to the condenser-pressure.

At the mean speed of 1,600 revolutions per minute, the tur-

* *Note on an accumulator-generator of steam*, by M. A. Rateau, Paris, 1902.

Utilization des vapeurs d'echappement par l'emploi d'accumulators de vapeur et de turbines à condensation. Bulletin de la Société de l'Industrie Minérale, premiere livraison, 1903.

Utilization des vapeurs d'echappement. Note publiée par la Société d'Exploitation des Apparails Rateau (accumulateurs de vapeur), 28 avenue de Suffren, Paris, Novembre, 1903.

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bine works two dynamos placed at the end of the shaft, and is sufficient for the service of pit No. 5 of these mines.

When the pumping engine is not in use, steam is furnished to the turbine by the boiler, and is expanded from its entrance by means of an automatic expanding arrangement.

In Figure 28 are the curves of the economy of this turbo-dynamo in function of speed.

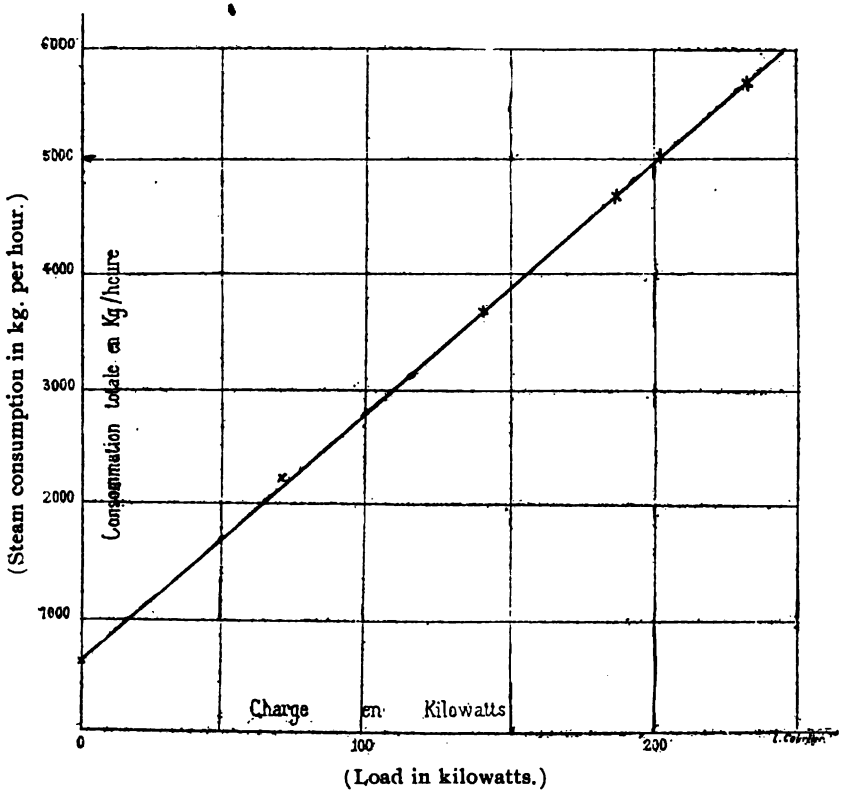

Fig. 28.—Low Pressure Turbo-Dynamo of 300 Horsepowers.

The figures which have served as basis for these curves, were obtained in the workshops of Sautter, Harlé and Co., by Messieurs Sauvage and Picou with every possible precaution.

The ensemble of the installation made in the mines of Bruay, is represented in Figure 29.

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There is shown behind the turbine, the steam-accumulator which regulates the flow of the exhaust steam delivered by the pumping engine.

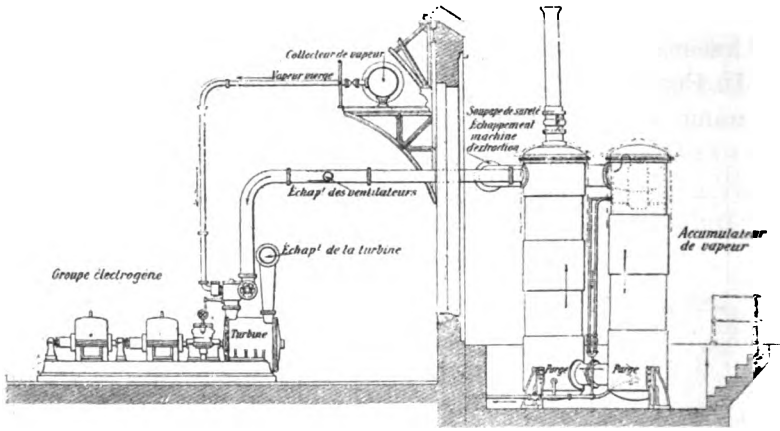

Fig. 29.

The following table gives the comparative results as calculated by the builders, and as obtained experimentally from the group in the mines of Bruay.

LOW-PRESSURE TURBO-DYNAMO OF 300 ELECTRICAL HORSEPOWERS.—RATEAU SYSTEM.

—	Calculated by the builders.	Results experimentally obtained.
Steam pressure, in pounds per square inch above zero	12.801	14.365
Steam pressure in the condenser, in pounds per square inch above zero	2.133	2.617
Revolutions per minute	1,800.000	1,800.000
Electrical horsepowers	300.000	336.000
Total net economy	0.570	0.580
Consumption of steam, in pounds weight per hour per electrical horsepower	38.360	37.368
Economy of the dynamos, measured by the method of opposition	.....	0.925

No more need be said regarding the innumerable applications that can be made of steam-turbines for the utilization of exhaust steam. There has been simply shown that the

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employment of multicellular turbines allows the combination of machines able to utilize alternately or simultaneously two flows of steam; the one of low pressure, the other of high pressure, both in the best conditions of economy. A turbine could thus be supplied by the exhaust steam from a rolling mill or any other primary engine, or by live steam drawn from a boiler, and in any proportion whatever, and for any length of time desired.

V.—APPLICATIONS OF THE STEAM-TURBINE TO THE PROPULSION OF VESSELS.

The application of the steam-turbine to the propulsion of vessels has caused, during late years, numerous discussions.

Mr. Parsons has the honor of being the first who propelled vessels entirely by turbines. His first attempt, *The Turbinia*, was soon followed by other vessels of light hulls and high speeds constructed for the British navy, and, more recently, by the *Queen* and by the *Brighton*, packet boats between England and France.

It is impossible to here treat in a complete manner problems which required a long development. The purpose is to only set forth certain consequences of the experience thus far acquired.

These are stated in a communication recently made by Mr. Rateau to the British Institution of Naval Architects, in which he gives some inedited notes regarding the last experiments in propulsion made with Rateau turbines.

DIFFICULTIES IN THE APPLICATION OF TURBINES TO THE PROPULSION OF VESSELS.

Notwithstanding that, to a first view, the replacing of piston steam-engines by steam-turbines for the propulsion of vessels seems easy, and that the advantages of the change appears evident, a more profound study of the case discovers difficulties under the three following heads:

1st. The adjustment of the screw propellers to the great rotary speed of the turbines.

2d. The obtaining of a good economy when the power is reduced for reduced speeds of vessel.

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3d. For performing the backing and maneuvering of the vessel.

ADJUSTMENT OF THE SCREWS TO THE GREAT ROTARY SPEED OF THE STEAM-TURBINE.

When the turbines are allowed to revolve at the speed which gives their maximum economy, they give excellent economic results, certainly better than can be obtained from the best piston steam-engines.

The figures that have been previously given for different applications of turbines, confirm this fact; as do, also, the published results of the experiments made with the turbines of Parsons.

Unfortunately, the rotary speed which assures a good economy with turbines, is too high for adaptation to helicoidal propellers, and the sacrifice, in part, of that economy becomes necessary if grave errors with the screws are to be avoided.

In order to turn the turbine slower than would be necessary with other applications of it, the number of working wheels must be increased, thus dividing the turbine into several parts, the steam cascading from the first part to the last part. On the other hand, in order to permit the screws to have a corresponding speed of rotation, they must be placed one, two or three on each of several shafts, and their surface must be so proportioned that their peripheral diameter is greater than their pitch.

These modifications diminish the economy of the screws, as well as they diminish the economy of the turbine; and there cannot be shown *a priori* that the mean economy of turbine and of screws is superior to that of a screw worked by a piston steam-engine.

The difficulties increase for slower speeds of vessel. This results, on the one part, from the fact that the total surface and, consequently, the dimensions of the screws, are determined by the greatest immersed transverse section of the vessel, while, on the other part, the size of the turbine is determined solely by the speed of its rotation, and not in the least respect by its power.

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In measure as the speed of the vessel is reduced, the speed of the turbine must be proportionally reduced, so that the dimensions of the turbine increase either by the diameter of the wheels or by their number during the diminution of its power, which latter is approximately in the inverse ratio of the cubes of the speeds. There is then a limit of speed below which turbines cannot be practically employed.

Mr. Rateau has expressed the opinion that this limit is probably about 20 sea-miles per hour. For very large vessels and consequently, very large powers to be developed, this figure, perhaps, might be a little reduced; but there is a very small probability that it could be much lowered.

ECONOMY AT LOW SPEEDS.

If the steam-turbine is capable of giving good results as regards economy at maximum power, its economy at reduced speeds becomes, frankly, very bad, because of the diminution of the hydraulic economy as has previously been shown.

At reduced speed the expenditure of steam by a turbine is always more than by a piston steam-engine, but at normal speed this expenditure can be a great deal less.

This disadvantage would not be of much importance for such vessels as passenger-steamers which nearly always go at maximum speed, but, for war vessels, whose most economical speed is far less than their maximum speed, which indeed is only exceptionably employed, there results that their increased consumption due to the decreased economy, enormously diminishes their radius of action.

Parsons has proposed to remedy this defect by adding a supplementary turbine called the "cruising turbine" into which the steam is first introduced when the steaming is done at a low speed, nevertheless, this process does not improve the hydraulic economy, and the consumption of steam remains high.

The only efficacious way for obtaining an economic functionment when steaming at low speeds, is to employ, in conjunction with the turbine, a piston steam-engine of less power.

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BACKING AND MANEUVERING.

With piston steam-engines, stopping and reversing are done in the simplest manner.

Turbines, on the contrary, are from their very nature irreversible.

Different inventors have proposed special arrangements of the bucketing allowing the same wheels to operate in both the forward and the backward direction. These attempts are not destined to succeed, because they can effect reversibility only by sacrificing the economy in the forward direction. Consequently, for backing, a piston steam-engine must be employed, or an additional steam-turbine.

But, unfortunately, the employment of a special turbine, has the disadvantage of greatly increasing the length of the space occupied by the machinery.

Mr. Rateau patented in 1898 the ingenious idea of placing in the interior of a principal turbine, and on its low pressure side, a backing turbine composed of several wheels with reversed bucketing, receiving steam from a special distributor.

This arrangement occupies less space than a separate turbine, and it is not detrimental when the vessel is going forward, as the backing wheels do not create any sensible resistance when the principal motor functions. It has been applied to the torpedo-boat *243* and to the *Libellule*.

But under any circumstances, it is difficult with turbines, to obtain a great backing speed. For that purpose, the engine must be able to develop as much power in backing as in going ahead; this is a serious disadvantage even with piston steam-engines and they are absolutely reversible.

As to the maneuvering in coming alongside of vessels, or of docks, the prompt stopping of the turbines has some difficulties:

Thus, when the steam-admission valve is closed, the screws continue to turn under the impulsion of the water, and easily entrain the wheels of the turbine, because their resistance to the movement of rotation is very feeble.

Also, the stopping and reversing, and, in general, the ease

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and rapidity for maneuvering the vessel, are very serious difficulties with the steam-turbine, and especially for vessels of war.

APPLICATIONS THAT HAVE BEEN MADE TO VESSELS.

FRENCH TORPEDO BOAT 243.

In 1898 the French Navy began to experiment with steam-turbines for the propulsion of vessels of war.

That Navy ordered from the Forges et Chantiers de la Méditerranée the experimental torpedo boat *243* of 92 tons, furnished with Rateau turbines.

This vessel was provided with two screw-shafts on each of which was a turbine in which the steam was completely expanded from the pressure in the boiler to the pressure in the condenser.

Unfortunately, the programme adopted was to utilize the hull of an ordinary torpedo-boat, and of placing the turbines in the space habitually allotted to its piston steam-engine.

That programme caused much inconvenience and embarrassment in the installation of the turbines and screws, and it prevented the experiment from being as conclusive as it was intended to have been.

In fact, there is much difficulty in adapting turbines to a hull originally designed to receive the shaft of a single piston steam-engine. The form of the hull not being sufficiently flat in its lower part, there was an impossibility of lowering the shafts enough at the place where they left the turbines, so that these shafts had to be much inclined—about 11 degrees.

Each shaft carried three screws, and was held by brackets secured to the hull. These brackets had a considerable length, and a resistance that could not have been negligible.

Figure 23, Plate VI, shows the stern of the torpedo-boat with the screws.

The conditions of the installation were very unfavorable. The much inclination of the shafts rendered the economical performance of the screws quite defective, as was ascertained from various and prolonged experiments.

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The turbines functioned in a very satisfactory manner, as stated in all the proces-verbaux of the experiments.

Below will be found copies of two of these proces-verbaux, in which are given the different speeds of vessel obtained with different pressures, and all the data necessary for valuing the qualities of the screws employed.

The two turbines are nominally of 900 horsepowers, and each is completely independent of the other. They contain at the exhaust end a single moving wheel for backing, which maneuver is consequently voluntarily sacrificed.

The speed of vessel obtained under the best conditions, has slightly exceeded 21 nautical miles per hour, while the speed which should have been obtained is about 24 nautical miles per hour, if the screws had given the same economy as those employed with the piston steam-engine at reduced speeds.

The screws employed with the turbine were much modified during the experiments. The number experimented with on each shaft was one, two and three, with very variable ratios of their pitch to their diameter, and having resistances proportionally variable.

The general conclusion that can be drawn from the experiments is the necessity of avoiding the inclination of the shafts, and the necessity of constructing the hull with proper adaptation for propulsion by turbines.

The turbine employed was of the kind first designed by the inventor, and had inside pillow-blocks. A flow of oil under pressure had been provided at the place where the shaft traverses the end of the turbine on its low pressure side.

An improper detail regarding this lubrication caused some trouble, the joint hindered the entrance of air, and entrained the oil into the condenser.

By the aid of a new system, entirely tight, it has been found possible to suppress even the low pressure pillow-block of the turbine itself, and to completely correct the trouble stated.

These details are not useless, because of many rumors concerning the prolonged experiments with the torpedo-boat *243*, attributing the speed failure to the bad functioning of the turbines.

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The experiments have been extended through the years 1902 and 1903, and will soon be completed. The French Navy has desired to ascertain as completely as possible the difficulties of the problem of screws worked at great rotary speeds, and the means of overcoming them.

Up to the present date, not less than seven systems of screws have been experimented with, designed by different specialists, and combined in various groupings.

Figure 24 Plate VI, represents the torpedo boat *243* under way.

LIBELLULE.

Another vessel of less tonnage—a vedette-torpedo boat—has been constructed for the French Navy with its turbine in two bodies instead of in one body as the turbine of *243* was, but with analogous power.

This vessel has only a single shaft. The experiments with it have not yet been commenced. The boilers are of a new system, and will be tried for the first time in the *Libellule;* the experiments with them are not yet completed.

BOAT BY YARROW AND COMPANY.

Messieurs Yarrow and Company have recently constructed, with a Rateau turbine, a light boat of the same type as the *Tarantula* built by the same company and furnished with a Parsons turbine.

This type of boat with the exception of the screws, is identical with the first class torpedo-boats of the British Navy. Its displacement is 140 tons and its length is 151 feet.

Its boilers, of the well known Yarrow type, have given the vessel by means of a piston steam-engine, a speed of about 27 geographical miles per hour.

This vessel is furnished with three shafts each carrying screws. The central shaft is worked by a piston steam-engine of 250 effective horsepowers; and this engine is entirely independent of the turbines.

The side shafts are arranged to receive for each of them one or two screws, and are each operated by a turbine in two bodies or in "cascade."

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FRENCH TORPEDO-BOAT No. 248.

Draught of Water { Forward, 3.6418 feet. Aft, 4.4948 feet. Mean, 4.0683 feet. Difference, 0.8530 feet.

Screws *F C M* { Number of blades in each screw, 4. Diameter of the screw, 1.96854 feet. Pitch of the screw, 1.63750 feet. Fraction of the pitch per blade, 0.11000.

EXPERIMENTS ON THE CONSUMPTION AT 20 KNOTS MADE JANUARY 22, 1903.

	Times of observation.	Direction of the base.	Time of making the run, in minutes and seconds.	Number of revolutions. By the counter. Starboard.	Port.	Means per minute.	Speed in knots.	Advance of vessel in feet per revolution.	Observed pressures by manometers, in pounds per square inch above the atmosphere. At the pumping plant.	Boilers. Forward.	Aft.	Collector of the steam.	Turbines. Starboard.	Port.	Pressure in condenser in pounds per square inch above zero.	Feed water heater.	Air pressure in boiler room, in pounds per square inch above the atmosphere.	Revolutions of the blowing engines per minute.	Fraction of the steam valve opening used. [Steam throttling at the turbines.]	Temperatures in Fahr. deg. Feed water.	Engine room.	Revolutions per minute of the steam engines driving the pumping plant.
	2 h. 17 m.	Beginning.		...	...	...	...	...	216.2	209.1	211.9	209.1	133.7	126.6	1.354	0.013	1.741	260	0.375	201.2	80.6	66
a	2 h. 30 m.	N. S.	3′-13″	5,490	5,600	1,723.83	20.438	1.199603	217.6	211.9	213.3	210.5	133.7	126.6	1.354	0.015	1.741	275	0.375	206.6	82.4	66
b	2 h. 42 m.	S. N.	3′- 3″	5,320	5,450	1,765.57	21.554	1.235265	219.0	213.3	214.8	213.3	135.1	128.0	1.354	0.015	1.741	265	0.375	204 8	80.6	68
c	2 h. 56 m.	N. S.	3′-15″	5,590	5,700	1,738.46	20.228	1.177326	217.3	210.5	213.3	210.5	130.9	123.7	1.335	0.015	1.934	240	0.375	203.0	82.4	68
I		Means = $\frac{a+b+b+c}{4}$ =				1,748.36	20.9435	1.211873	Slip of screws = 0.260.													
	3 h. 17 m.	Free route.		...	...	...	...	...	216.2	210.5	211 9	209.1	133.7	126.6	1.335	0.013	1.837	250	0.375	204 8	80.6	66
d	3 h. 30 m.	S. N.	2′-56½″	5,150	5,290	1,774.50	22.349	1.273421	230.4	221.9	226.1	220.5	135.1	128.0	1.335	0.015	1.934	270	0.375	204.8	82.4	67
e	3 h. 42 m.	N. S.	3′-17″	5,740	5,860	1,766.50	20.023	1.146880	229.0	220.5	223.3	219.0	133.7	126.6	1.335	0.015	1.934	265	0.375	206.6	82.4	69
f	3 h. 54 m.	S. N.	2′-54″	5,140	5,250	1,791 38	22.669	1.280442	227.6	221.9	223.3	214.8	138.0	130.8	1.335	0.015	1.934	265	0.375	206.6	80.6	67
II		Means = $\frac{d+e+e+f}{4}$ =				1,774.72	21 266	1.212135	Slip of screws = 0.260.													
		Means = $\frac{I+II}{2}$ =				1,761.54	21.105	1.212004	Slip of screws = 0.260.													
	4 h. 17 m.	End.		...	...	...	...	...	227.6	221.9	223.3	220.5	133.7	126.6	1.373	0.015	1.934	265	0.375	204.8	80.6	66

OBSERVATIONS.—Total grate surface, 44 9 square feet. Duration of experiment, 2 hours. *Consumption of Coal:* Total pounds, 7,231.0; pounds per hour, 3,615.5; pounds per hour per square foot of grate, 80.523; pounds per hour per mile (20.972), 142.4. *Consumption of Oil:* Total olive oil, in pounds, 39.68; total mineral oil, in pounds, 13.22; aggregate pounds, 52.90. The general functionment has been very satisfactory.

Total number of revolutions during the experiment, starboard, 208,100; port, 212,740.

Mean number of revolutions made per minute, starboard, 1,734.17; port, 1,772.83.

Mean number of revolutions made per minute for the two turbines, 1,753.50.

Mean advance of the vessel in feet per revolution = 1.212004.

Speed of the vessel in knots = $\frac{1,753.5 \times 1\ 212004}{101.33664}$ = 20.972.

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FRENCH TORPEDO-BOAT No. 248.

Draught of water { Forward, 3.6909 feet. Aft, 4.4445 feet. Mean, 4.0677 feet. Difference, 0.7536 feet.

Screws *F C M* modified { Number of blades in each screw, 4. Diameter of the screw, 1.96854 feet. Pitch of the screw, 1.63943 feet. Fraction of the pitch per blade, 0.11060.

EXPERIMENTS MADE SEPTEMBER 17, 1908. (SCREWS *AV* AND *M* IN PLACE.)

Times of observation.	Direction of the base.	Time of making the run, in minutes and seconds.	Number of revolutions. By the counter. Starboard.	Port.	Means per minute.	Speed in knots.	Advance of vessel in feet per revolution.	Observed pressures by manometers, in pounds per square inch above the atmosphere. At the pumping plant.	Boilers. Forward.	Aft.	Collector of the steam.	Turbines, mean of starboard and port.	Pressure in condenser in pounds per square inch above zero.	Feed water heater.	Air pressure in boiler room, in pounds per square inch above atmosphere.	Revolutions of the blowing engines per minute.	Fractions of the steam valve opening used. [Steam throttling at the turbines.]	Temperature in Fahrenheit degrees of the feed water.	Revolutions per minute of the steam-engines driving the pumping plant.
2 h. 57 m.	DA	5′-48″	6,870	7,100	1,204.3	14.054	1.181755	210.5	196.3	203.4	200.5	42.67	1.315	—5.80	0.484	242	0.125	188.6	40
3 h. 12 m.	AD	5′-29″	6,610	6,820	1,224.6	14.866	1.229327	207.7	197.7	200.5	197.7	41.25	1.325	—3.87	0.387	226	0.125	188.6	40
			Means,		1,214.4	14.460	1.205541	Slip of screws = 0.2640.											
3 h. 30 m.	DA	4′-54″	7,000	7,300	1,459.1	16.635	1.154524	206.3	199.1	199.1	199.1	69.69	1.325	+1.93	0.677	240	0.250	201.2	40
3 h. 40 m.	AD	4′-54″	7,090	7,410	1,479.5	16.635	1.138448	210.5	199.1	203.4	202.0	72.54	1.325	+1.93	0.677	242	0.250	202.1	40
			Means,		1,469.3	16.635	1.146486	Slip of screws = 0.3006.											
3 h. 56 m.	DA	4′-21″	7,300	7,760	1,731.0	18.739	1.096125	207.7	199.1	200.5	199.1	99.56	1.315	+7.74	1.161	250	0.333	208.9	52
4 h. 8 m.	AD	4′-31″	7,680	8,160	1,753.5	18.047	1.042320	214.8	206.3	207.7	206.3	100.98	1.315	+7.74	1.161	252	0.333	207.5	56
4 h. 20 m.	DA	4′-15″	7,230	7,680	1,754.1	19.180	1.107280	214.8	206.3	207.7	203.4	102.40	1.315	+7.74	1.354	252	0.333	210.2	56
			Means,		1,748.0	18.503	1.072011	Slip of screws = 0.3462.											
4 h. 33 m.	AD	4′-18″	8,310	8,760	1,984.9	18.957	0.967189	236.1	227.6	229.0	226.1	128.00	1.296	+15.44	1.741	280	0.375	213.8	60
4 h. 43 m.	DA	3′-55″	7,630	8,000	1,995.3	20.812	1.056099	227.6	220.5	223.3	217.6	124.45	1.296	+13.54	1.934	280	0.375	212.0	60
4 h. 55 m.	AD	4′-18″	8,480	8,940	2,025.5	18.957	0.947504	226.1	219.0	216.2	214.8	126.58	1.296	+15.44	1.741	280	0.375	210.2	60
			Means,		2,000.2	19.8845	1.006723	Slip of screws = 0.3860.											

OBSERVATIONS.—Started blowers at 1 h. 55 m. Equalized the turbines at 2 h. 10 m. Began the experiments at 2 h. 30 m. Ended the experiments at 5 h. 20 m. Smooth water. Light breeze from E. to S. Barometer 30.31 inches. Consumed 28 quarts of olive oil. Consumed 4 quarts of mineral oil. The general functionment has been very satisfactory.

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The combination of a piston steam-engine and steam-turbines adopted by Messieurs Yarrow & Co., had been previously suggested by Mr. Soliani, director of the Ansaldo workshops. The combination can be made in various ways.

Mr. Rateau has shown a way of combining the piston steam-engine with steam-turbines, by directing its exhaust steam into either the high pressure or into the low pressure turbine, according to the case.

Without entering into details, the results of the experiments show that the turbines make 2,000 revolutions per minute (they were calculated for 1,500 revolutions), and that they develop over 2,000 horsepowers with a consumption of a little more than 8.8 pounds of steam per hour per horsepower.

The maximum speed attained during the experiments which have been made since October last, is 26.25 geographical miles per hour. Several different screws have been tried with one screw and two screws on each shaft.

The opinion of Mr. Yarrow's engineers is, that it would be possible to attain, if not the speed of 27 geographical miles per hour, at least a speed not much inferior.

As regards the mechanical or practical working, the Rateau turbine in the Yarrow boat showed itself an excellent machine, as did also the Rateau turbine in *243*.

There was no accident of any kind, neither rupture of a blade, nor interference of mechanism, nor vibrations, and the admission is now generally made that the multicellular turbine is not inferior in any respect to any other system known on board of vessels.

Figure 25 Plate VI, gives a general view of the vessel constructed by Messrs. Yarrow & Co.

RESUMÉ AND CONCLUSIONS.

To sum up: there has been given in this work a general classification of the different systems of steam-turbines, a classification in which has been included not only the systems actually known to, and employed in, practice; but, also, those which could be formed by the application of the general principles of mechanics to this kind of machine.

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There has been pointed out the place which the Rateau turbine occupies in such a classification, and from it the immediate result of the advantages of the multicellular system of action.

The question of economy has then been investigated by analyzing the different elements on which it depends, that is to say, first, the hydraulic economy; the losses occasioned by the leakage of steam, and by its frictions and eddies.

Then, this analysis has been applied to the comparison of the two turbines of action, namely, the multicellular turbine with pressures in stages, and the turbine having groups of wheels with velocities in stages.

This discussion has brought forward in a complete manner the economic superiority of the first combination.

Divers combinations recently announced as new systems have been examined, and their economic and mechanical performances determined.

The purpose of the second part of this work has been the exposition of the applications which have been made of the Rateau turbine; first, to the generation of electricity; and, afterwards, to the turbo-pumps and to the turbo-ventilators; then, to the utilization of exhaust steam; and, finally, to the propulsion of vessels.

It is hoped that this review, sometimes summary, will suffice to show the general interest that attaches to the system invented by Mr. Rateau, an interest so much the greater as that engineer has known how to create the necessary methods for calculating the turbines in all the various conditions of their application with a rigor proved by the preceding examples, and which has not been attained by even the piston steam-engines.

In conclusion, the remark seems proper that if the honor of the first industrial turbine belongs to England in the person of Mr. Parsons, the labors of Mr. Rateau and of the firm of Sautter, Harlé and Company, assures, nevertheless, on their part for France, a marked place in the development of the new steam-motor.

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TORPEDO BOAT No. 20, *GOLDSBOROUGH.*

DESCRIPTION AND OFFICIAL TRIAL PERFORMANCE.

By G. H. Thayer, Draughtsman, U. S. N.

The *Goldsborough* (built by Wolff and Zwicker Iron Works, Portland, Oregon) was one of three torpedo boats authorized by the Act of March 3, 1897.

The contract was signed July 30, 1897, the price being $214,500 (exclusive of ordnance and equipment fittings, which were supplied by the Government). The time allowance for completion was eighteen months.

The speed guaranteed in the original contract was 30 knots per hour, to be maintained for two consecutive hours over a course not less than thirty nautical miles in length. The vessel to carry the following weights additional to that of the complete hull and machinery, with spare parts and the water necessary for the trial, viz: Torpedo tubes, gear and torpedoes, rapid-fire guns and ammunition, 12.6 tons; coal, 20 tons; crew, boats, anchors, electric plant, equipment, etc., 9 tons.

After a long series of trials, accidents and delays, the Navy Department (January 21, 1902) declared the contract forfeited on the part of the contractors. On March 3, 1905, an Act of Congress was approved authorizing the Secretary of the Navy to accept the *Stringham* and *Goldsborough* after a satisfactory trial at reduced speed. It was decided to run the standardizing trial up to and including 27 knots, and the one-hour's run not to exceed 27 knots.

HULL DATA.

Length between perpendiculars, feet and inches	198– 0
L.W.L., feet and inches	198– 0
Beam, extreme, feet and inches	20– 7¼
L.W.L., feet and inches	19–11

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Depth, M.S., feet and inches	12– 6
Mean draught on trial, feet and inches	6–10$\frac{5}{16}$
Displacement on trial, tons	255.5
full load, tons	360
per inch, L.W.L., tons	6.3
Area of immersed M.S., square feet	76
L.W. plane, square feet	2,670
Center of gravity of L.W. plane aft of M.S., feet and inches	7– 9
buoyancy above keel, feet	3.6
Transverse metacenter above center of buoyancy, feet and inches	6–11
Longitudinal metacenter above center of buoyancy, feet	595
Number of frames	117
Watertight compartments	22

HULL CONSTRUCTION.

The hull is constructed of mild steel of domestic manufacture and galvanized to two feet above the load water line.

The frames are spaced 20 inches between centers and are $2\frac{1}{4}$ inches × $1\frac{1}{2}$ inches × 2.3 pounds angle steel amidships and 2 inches × $1\frac{3}{8}$ inches × 2.1 pounds angle steel at ends of ship.

Reverse frames are 2 inches × $1\frac{3}{8}$ inches × 2.1 pounds angle steel. There are eight belt frames, 6 inches deep, in engine space.

The flat keel is 10-pound plate amidship, reduced to 9-pound and 8-pound plates at end.

The center-line keelson is $7\frac{1}{2}$-pound plate in engine and boiler rooms, and intercostal 5-pound plate at ends.

The shell plating is 6-pound plate amidship for half the length of the ship, reduced to 4.43 pounds at ends, except garboard and sheer strakes, which are—garboard, 8-pound to $6\frac{1}{2}$-pound; sheer strake, $8\frac{1}{2}$-pound to 5-pound.

The watertight bulkheads, of which there are eleven, extending to the deck, divide the ship into twelve main compartments. The forepeak, storeroom and capstan room, machinists' room, with locker, galley, athwartship coal bunker, forward boiler room, after boiler room, engine room, crew's quarters, officers' quarters, officers' toilet and pantry and storeroom in the stern.

Magazine and storerooms are provided below crew's and machinists' quarters.

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The bridge is located on top of the forward conning tower, and is fitted with the steering wheel controlling the steering engine, which is located in the engine room; a tell-tale, indicating the position of the tiller, engine-room indicators and voice tube to engine room.

The forward conning tower, which is of 7½-pound steel plate, contains steering wheel, voice tube to engine room, chart locker and entrance hatch to machinists' quarters.

The after conning tower, which is also of 7½-pound steel plate, contains a hand steering wheel, engine-room indicators and entrance hatch to crew's quarters.

The boats, two of which are 20 feet long, of Navy model, and one of the canvas folding type, are carried on deck amidships.

Two anchors of the Baldt stockless type are carried, each stowed on skids, on the sides of turtle deck. These anchors weigh about 625 pounds each.

ARMAMENT.

The battery consists of four 6-pounder semi-automatic rapid-fire guns, mounted on pedestal mounts with hydraulic recoils. These guns are located as follows: One on the starboard side, over the forward fireroom coal bunker; one on the port side, over the after fireroom coal bunker; one on the starboard side, over the crew's quarters; one on the port side over the officers' quarters. This location gives all the guns the maximum amount of train combined with efficiency.

Torpedo tubes, of which there are two, located on the center line of the ship, train on either beam. The forward tube is located over the after part of the engine room and the after tube is located aft on the stern and over the after part of the officers' quarters. Four modern Whitehead torpedoes of the latest pattern are carried, two in the tubes and two under the turtleback forward.

MACHINERY.

Main Engines.—There are two triple-expansion, four-cylinder engines, of the vertical, inverted type, placed back to back in the same engine room, and designed to develop a total of

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6,000 horsepower at 360 revolutions per minute. The engines are right and left, turn outboard, and drive twin screws.

The arrangement of the cylinders is: High pressure, intermediate pressure, forward low pressure and after low pressure. The cranks are at 90 degrees, the H.P. and I.P. cranks being opposite, as are also the 1st L.P. and 2d L.P. cranks.

The H.P. and I.P. cylinders are in one casting, as is also the case with the two L.P. cylinders; the two castings are bolted together and supported on a framing of nickel-steel vertical columns well stayed with diagonal braces.

The cylinder covers are composition, cast with stiffening ribs, and contain the steam ports and passages; also the valve liners.

The main valves are all single-ported plug valves, of the piston type, one for the H.P. cylinder and two each for the I.P., 1st L.P. and 2d L.P. cylinders; these valves are actuated (through a rock shaft) by means of a valve gear of the Stephenson link type, with double bars. The link block is carried at the center of a vertical link, the lower end of which connects to an arm of the rock shaft, and the upper end to a vibrating lever hung from the cylinder cover. The rock shaft turns in bearings carried by the back frames and main guide castings, and operates the main valves by means of arms forged from the solid. Each valve stem and its crosshead is in one forging of nickel-steel. The eccentrics are of cast steel, in two parts, bolted together and keyed to raised seatings on the crank shafts. The eccentric straps are of composition, lined with white metal, and the eccentric rods are of forged steel. Adjustment of cut-off is effected by means of a sliding block in the reverse-shaft arm which is shifted by means of a threaded bolt.

CYLINDER DATA.

	H.P.	M.P.	Each. (2) L.P.
Diameter of cylinders, starboard, inches	19⅝	31¾	35¾
port, inches	19¼	31¼	35¾
Liners in H.P. and I.P.	—	—	—
Stroke, inches	20	20	20
Thickness of bonnets, starboard, inch	⅞	⅞	⅝
port, inch	⅝	⅝	⅝

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	H.P.	M.P	Each. (2) L.P.
Number of valves	1	2	2
Diameter of valves, inches	$12\frac{15}{16}$	$10\frac{15}{16}$	$10\frac{15}{16}$
valve stem, inches	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$
Side of valve on which steam is taken	inside	outside	1st inside 2d outside
Valve travel, inches	$4\frac{7}{8}$	$5\frac{1}{4}$	$5\frac{1}{4}$
Steam lead, top, inch	$\frac{13}{32}$	$\frac{15}{32}$	$\frac{17}{32}$
bottom, inch	$\frac{17}{32}$	$\frac{31}{32}$	$\frac{1}{2}$
Cut-off, top, inches	$14\frac{1}{4}$	$14\frac{1}{4}$	$14\frac{1}{4}$
bottom, inches	$13\frac{1}{4}$	13	$13\frac{1}{4}$
Exhaust, top, inches	$17\frac{1}{2}$	$17\frac{5}{8}$	$17\frac{5}{8}$
bottom, inches	$17\frac{5}{8}$	$17\frac{1}{2}$	$17\frac{1}{2}$

The main pistons are conical, of cast steel, and fitted with two hard cast-iron snap rings. Each piston rod is hollow and, with its crosshead and slipper, is in one piece forged of nickel-steel and oil tempered, the slipper being faced with white metal. The piston rods are secured to the piston by means of a taper fit and a nut.

PISTON RODS.

	H.P.	I.P.	Each. L P.
Diameter of piston rod, inches	4	4	4
axial hole, inches	$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{5}{8}$
Width of crosshead slipper, inches	12	12	10
Length of crosshead slipper, inches	14	14	14
Width (total) backing surface, inches	$5\frac{1}{2}$	$5\frac{1}{2}$	$5\frac{1}{2}$

The crosshead guides, of the slipper type, are of cast iron, hollow, for water circulation. The upper ends are bolted to lugs cast on the lower cylinder covers, and the lower ends bolted to forged-steel strongbacks, which are, in turn, secured to the engine columns. The backing guides are of cast iron and bolted to the face of the go-ahead guides.

The connecting rods are hollow. They are of forged nickel-steel, oil-tempered. The crosshead pins are hollow, of oil-tempered nickel-steel, and are shrunk into the eyes of the connecting rods. The hollow space which is used as an oil reservoir, communicates by means of a small hole with the upper surface of the pin. The crankpin caps are of composition, and they, with the lower end of the connecting rods, are lined with white metal. All crank-pin and crosshead-pin bolts are of nickel-steel.

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CONNECTING RODS.

Length of connecting rods, center to center, inches	44
Diameter at neck, inches	3⅝
bottom end, inches	4¼
of axial hole, inches	2
crosshead pin, inches	5
Length of crosshead pin, inches	7
Diameter of axial hole, inches	2½
crosshead-pin bolt, inches	2
crank-pin bolts, inches	2⅛

Each crank shaft is in one section, hollow, of forged nickel-steel, oil-tempered. The cranks are at 90 degrees. The axial hole in the crank pin is utilized as an oil reservoir, being connected by a small hole to a centrifugal oiler, fastened to the side of the crank web, and having a small hole communicating with the outside surface of the crank pin, through which the oil is thrown by centrifugal force.

CRANK-SHAFT DATA.

Crank pin, diameter, inches	7⅛
length, inches	10
Diameter axial hole, inches	3½
of crank shaft, inches	7⅛
axial hole, inches	3½
eccentric seating, inches	7⅝
coupling flange, inches	13½
Thickness of coupling flange, inches	1½
Diameter of coupling bolts, inches	1 21/32
Crank webs, width, inches	8½ and 7⅝
thickness, inches	4
Diameter of main bearings, inches	7⅛
Length of main bearings, one H.P., I.P., inches	8
four H.P., I.P., inches	12
Diameter of main bearing bolts (2), inches	2

The engine bed consists of five main-bearing bridges, bolted to two steel angle bars by body-bound bolts. These angles are seated on and bolted by fitted bolts to the main-engine keelsons. Each bridge has an extension or arm reaching down, through the end of which a longitudinal bolt passes under the engine, clearing the cranks and binding all the bridges together, thus overcoming the tendency to roll or twist laterally. The main-bearing bottom brasses are circular, to facilitate re-

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moval, and the caps are hollow, for water circulation; they are lined with white metal.

The thrust shaft, of oil-tempered nickel-steel, is in a short section connected to the crank and line shafts by means of flanges and taper bolts. It works in a thrust bearing of the ordinary horseshoe type, the body of the bearing is of composition so shaped as to form an oil reservoir, and has side lugs or brackets for bolting it between the main-engine foundation angles.

The horseshoes are of composition faced with white metal. Oil ways run to the working faces from a reservoir cored in the top and provided with wick feeds. The shoes are held in place by extension lugs fitting between lock nuts threaded on side adjusting rods extending the length of the box.

THRUST-SHAFT DATA.

Diameter of flange couplings, inches	13½
shaft, inches	7
axial hole, inches	4½
over collars, inches	11
Number of collars	7
Thickness of collars, inches	1⅛
Space between collars, inches	2½
Number of horseshoes	7
Diameter of side adjusting rods, inches	2
Length of thrust-shaft spring bearings, inches	7½
Diameter of thrust-shaft spring bearings, inches	7½

The line and propeller shafts are also of oil-tempered nickel-steel. The line shafting is connected to the thrust shaft by means of flanges, and to the propeller shaft by means of a taper and socket joint with cross key and feather. The shaft is covered with a sleeve where it passes through the stern tube. The after end of the propeller shaft is turned with a taper to fit the propeller boss.

LINE-SHAFT DATA.

Diameter of shaft, inches	7
axial hole, inches	4½
spring bearing, inches	7½
Length of spring bearing, inches	12½
Diameter over sleeve, inches	7½

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Diameter in stuffing box, inches	7¾
Length of strut bearing, inches	15⅝
Diameter of strut bearing, inches	7¾
Coupling sockets, outside diameter, inches	10¼
taper, per foot, inch	¾
Length of taper, inches	13¾
Cross keys, inches	3½×1
Feather key, inches	1¼× ¾

The propellers are three-bladed, cast solid, of manganese-bronze and polished. Each propeller is fitted to the taper on its shaft and secured by means of a feather key and a nut on the end of the shaft, and locked in place. The nut is threaded in the opposite direction to the rotation of the propeller when going ahead.

PROPELLER DATA.

Number of blades	3
Diameter, feet and inches	7– 6
Mean pitch, starboard, feet and inches	10– 0.18
port, feet and inches	10– 4.8
Ratio, pitch ÷ diameter	1.36
Helicoidal area, square feet	37
Projected area, square feet	29.04
Disc area, square feet	44.179
Ratio, projected area ÷ disc area	.656
Area of immersed midship section	78.4
Ratio disc area to area immersed midship section	.563
projected area to area immersed midship section	.37
Diameter of hub, inches	14
nut thread, inches	3

REVERSING ENGINE.

The reversing engine consists of an oscillating cylinder, with the trunnion plate bolted to the vertical web of the foundation angle bar between the I.P. and 1st L.P. engines.

The end of the piston rod is directly connected to an arm forged on the weight shaft. The valve of the reversing engine is of the common "D" type, actuated by the ordinary floating lever system. A hollow shaft extends to an operating hand lever, located near the main throttle valve at the front of the engine. Through this hollow shaft runs an extension stem to the steam valve, thus giving the machinist in charge admirable facilities for handling his machinery. The steam and

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exhaust pipes of the reversing engine are connected through the trunnions. At the forward end of the engine is also located a hand wheel, which can be thrown into gear with the weight shaft, to reverse the engine in case of an accident to the steam gear.

REVERSING-ENGINE DATA.

Diameter of steam cylinder, inches	4
Stroke of piston, inches	$9\frac{11}{32}$
Diameter of piston rod, inches	$1\frac{1}{4}$
connecting pin, inches	$1\frac{3}{8}$
weight shaft, inches	$3\frac{1}{4}$
hand reverse wheel, inches	24
Number of turns of wheel to reverse	$1\frac{1}{4}$

The turning gear, at the aft end of the engine, consists of a worm turned by a ratchet lever that can be thrown in mesh with a worm gear keyed on the flanges connecting the crank and thrust shafts.

TURNING-GEAR DATA.

Number of teeth in worm wheel	58
Pitch of worm, inches	1.245
Length of ratchet handle, inches	24

The condenser, into which both engines exhaust, is located on the center line of the ship, between the engines. The shell, which is of circular section, is formed of copper stiffened by circumferential corrugations. The condenser is curved in a vertical plane, the tubes being bent to concentric curvature and the ends expanded in the composition tube sheets, thus dispensing with ferrules and packing, the curvature of the tubes allowing for expansion from heat. The water chests are formed of sheet copper, the after one containing the propeller, by means of which water is drawn or forced through the tubes when the ship is at rest or under slow speed. There are two openings through the plate, with strainers and scoops, to force the water through the condenser when the vessel is under way.

CONDENSER DATA.

Diameter of shell, inside, feet and inches	3- 6
Thickness of shell	B.W.G. No. 10.
Number of tubes	1,972

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Thickness of tubes	B.W.G. No. 18.
Average length of tubes, feet and inches	12- 9
Spacing of tubes, inch	$\frac{13}{16}$
Cooling surface, square feet	4,006.66
Ratio of cooling surface to I.H.P.	.67
Diameter of exhaust pipe (2), inches	$18\frac{1}{4}$
injection pipe, inches	18
air-pump suction (2), inches	7

The main air pumps are of the trunk-bucket type, with floating top and foot valve plate, and are located at the forward end of the engines, under the floor plates, hanging by side brackets to the main-engine foundation angles. The air pumps are driven by a crank shaft having a projection at the after end that dovetails into a recess in the forward end of the main-engine crank shaft, thus allowing a slight independent alignment of the two shafts.

MAIN AIR-PUMP DATA.

Diameter of bucket, inches	$20\frac{1}{2}$
Stroke of bucket, inches	$3\frac{1}{2}$
Diameter of trunk, inches	$6\frac{3}{4}$
Number of valves in bucket	20
foot plate	20
floating top	25
Diameter of valves, inches	$2\frac{1}{2}$
Lift of valves, inch	$\frac{1}{8}$
Length of connecting rod, inches	$13\frac{5}{8}$
Diameter of crank shaft, inches	$3\frac{3}{4}$
Trunk pin, inches	$2\frac{1}{2} \times 5\frac{5}{8}$
Crank pin, inches	$3\frac{3}{4} \times 4$
Crank-shaft bearings, inches	$3\frac{3}{4} \times 5\frac{1}{2}$
Suction pipe diameter, inches	7
Discharge pipe diameter, inches	7

The main circulating pump consists of a propeller, located in the after water way of the condenser, driven by an independent, vertical, inverted, single-cylinder, slide-valve engine. The cylinder of this engine is supported on light, turned, wrought-steel columns. The forward main bearing forms the thrust, the collars on the shaft running incased in the white metal of the bearing. The engine is reversed by means of a shifting eccentric.

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CIRCULATING-PUMP DATA.

Diameter of cylinder, inches	6
Stroke of piston, inches	5
Diameter of piston rod, inches	$1\frac{1}{8}$
Crosshead pin, inches	$1\frac{3}{4} \times 4\frac{1}{8}$
Axial hole in pin, inch	1
Crosshead slipper, inches	$3\frac{1}{8} \times 3\frac{7}{16}$
Length of connecting rod (center to center), inches	10
Crank pin, inches	$2 \times 3\frac{1}{8}$
Axial hole in pin, inches	$1\frac{1}{8}$
Crank webs, inches	$1 \times 2\frac{1}{4}$
shafts, diameter, inches	$1\frac{3}{4}$
shaft bearings, inches	$2\frac{1}{4} \times 1\frac{3}{4}$
thrust bearings, inches	$4\frac{1}{4} \times 5\frac{3}{4}$
Number of rings	2
Circulating propeller, diameter, inches	20
pitch, inches	18
revolutions per minute	300

STEAM PIPES.

Two main steam pipes, one on each side, lead aft to the engines, with a cross connection and stop valve between them forward of the engines. There is also a main stop valve on each main line at the engine-room bulkhead. The throttle valves are of the rotating type geared to hand wheels, with divisions on them to indicate the degree of opening.

The auxiliary steam pipe pierces the bulkhead on the starboard side, has a cross connection to the main steam line, and a branch leading to the by-pass valve connected to the L.P. receivers.

BOILERS.

There are three water-tube boilers of the Thornycroft "Daring" type, one of which is located in the forward boiler room and two in the after boiler room. Each boiler has two furnaces, with two fire doors to a furnace. The furnaces are lined with firebrick. The grate bars are of mild steel made up in sections of five bars to each section. The boiler casing is of thin galvanized iron, lined with asbestos millboard. There is one twin-spring safety valve and one sentinel valve to each boiler. The feed water discharges into the lower part of the steam drum through a Thornycroft feed-water

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regulator connected to each boiler. The boiler in the forward boiler room and the forward boiler in the after boiler room discharge their smoke into the same smoke pipe, which is located over the watertight bulkhead between them. The after boiler in the after boiler room has its own smoke pipe located forward of the engine-room hatch.

BOILER DATA.

Number of boilers	3
Steam pressure, pounds	250
Length over casings, feet and inches	10- 2
Width over casings, feet and inches	13- 7
Height ash pan to top of casing, feet and inches	10- 3
Steam drums, length, feet and inches	12-11 11/16
drum, diameter, inches	37
thickness, inch	7/8
Water drums, number for each boiler	3
Side drums (2), diameter, inches	6 1/2
thickness, inch	1/2
Center drum, diameter, inches	18
thickness, inch	5/8
Tubes, downflow, diameter, inches	3 3/8
thickness, inch	3/16
number to each boiler	14
generating, diameter, O. S. inches	1 1/8 and 1 1/4
thickness	No. 11, B.W.G.
number to each boiler	1,780
Baffle tubes, diameter, inches	1 1/8 × 1 1/4
thickness	No. 11, B.W.G.
number to one boiler	808
Furnaces, number to each boiler	2
doors to each boiler	4
width of grate, feet and inches	4- 9 5/8
length of grate, feet and inches	7- 6
Total heating surface of three boilers, square feet	13,500
grate surface of three boilers, square feet	216
Ratio grate surface to heating surface	1 : 62.5
Smoke pipes, height above grates, feet and inches	23- 6
area of forward pipe, square inches	2,501.82
after pipe, square inches	1,320.25
Ratio of stack area to G. S.	17.7 : 1
I.H.P. per square foot of grate surface, main engines	27.8

FORCED DRAFT.

The closed-fireroom system of forced draft is used, air being supplied by three fan blowers, built by the contractors from

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their own designs. The blower engines are direct-connected, of the two-cylinder, horizontal, opposed type, with cranks at 180 degrees, and piston valves. The working parts are enclosed in a sheet-iron casing to protect them from the dirt and coal dust.

One blower is located in the forward fireroom, on a shelf fastened to the forward athwartship bulkhead, and takes its air from an air box, over which is mounted the forward 48-inch copper cowl. In the after fireroom are located the other two blowers, on shelves fastened to the starboard and port coal-bunker bulkheads, and they take their air from an air box between them, over which is mounted the after 72-inch copper cowl.

BLOWER ENGINE DATA.

Diameter of cylinders, inches	$3\frac{1}{2}$
Stroke of pistons, inches	$3\frac{1}{2}$
Diameter of piston rods, inch	$\frac{5}{8}$
valves, inches	$1\frac{15}{16}$
Travel of valves, inches	1
Crosshead pins, inches	$1\frac{1}{8} \times 4\frac{3}{8}$
slippers, inches	$2\frac{3}{4} \times 4$
Length of connecting rod, center to center, inches	$8\frac{1}{2}$
Crank pins, inches	$2 \times 2\frac{1}{8}$
web, inches	$\frac{3}{4} \times 2\frac{1}{8}$
shaft diameter, inches	2
angle, degrees	180
shaft bearings, inches	$2 \times 2\frac{1}{8}$
Fan, diameter, inches	60
width at rim, inches	$9\frac{1}{2}$
diameter air duct, inches	$30\frac{1}{8}$
Revolutions per minute	800

FEED AND FILTER TANKS.

In the engine room, with its back against the forward bulkhead, is the feed and filter tank. The main air pumps discharge into the starboard and port sides near the top. There are two compartments, containing three baskets each, of filtering material, through which the water from the air pumps is delivered on its way to a compartment to which the feed-pump suctions are attached. This feed tank is supplied with an overflow leading to the main feed tanks, a vapor pipe to

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the atmosphere, a thermometer, a gauge glass and connections from the steam traps.

PUMPS.

The main feed pumps, of which there are two, are of the Blake simplex Admiralty type, bolted to the forward bulkhead of the engine room on the port and starboard sides, the hotwell being between them. They take their supply from the hotwell, from the feed tanks, and from the sea, and discharge into the main feed line to the boilers.

The auxiliary feed, fire and bilge pumps, of which there are two, of the Blake simplex Admiralty type, are located in the boiler rooms. The smaller one is fastened to the forward bulkhead of the forward fireroom on the starboard side, and the larger one is located between the two after boilers on the center line of the vessel. Each is connected to a suction manifold, with pipes leading from the sea, hotwell, reserve tanks and drainage system. They are also connected to a discharge manifold, from which pipes lead to the main feed, overboard, fire main and auxiliary feed, to the boilers in its own compartment.

A pump for draining the condenser while the main engines are idle is of the Blake simplex Admiralty type, bolted to the forward bulkhead of the engine room on the starboard side. It draws its supply from the lower part of the main air pumps and discharges into the hotwell.

A water-service pump of the same size and type as the condenser pump is located on the forward bulkhead of the engine room on the port side. It draws its supply from the sea, and delivers into the water-service pipes and also into the distiller circulating pipes.

PUMP DATA.

Main feed pumps (*2*).

Type	Vertical simplex.
Diameter of steam cylinder, inches	12
water cylinder, inches	8
Stroke of piston, inches	12
Gallons per double stroke	5.222
Diameter of steam pipe, inches	1½
exhaust pipe, inches	2½
suction pipe, inches	3½
discharge pipe, inches	2¾

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Auxiliary Pumps (2). After Boiler Room.

Type	Vertical simplex.
Diameter of steam cylinder, inches	12
water cylinder, inches	8
Stroke of piston, inches	12
Gallons per double stroke	5.222
Diameter of steam pipe, inches	2
exhaust pipe, inches	2½
suction pipe, inches	3½
discharge pipe, inches	2¾

Forward Boiler Room.

Type	Vertical simplex.
Diameter of steam cylinder, inches	10
water cylinder, inches	6
Stroke of piston, inches	12
Gallons per double stroke	2.938
Diameter of steam pipe, inches	1½
exhaust pipe, inches	1¾
suction pipe, inches	3
discharge pipe, inches	2½

Condenser Pump.

Type	Vertical simplex.
Diameter of steam cylinder, inches	3½
water cylinder, inches	4
Stroke of piston, inches	4
Gallons per double stroke	.4352
Diameter of steam pipe, inch	¼
exhaust pipe, inch	½
suction pipe, inches	2
discharge pipe, inches	1¾

Sanitary Pump.

Same as condenser pump.

Distiller Circulating Pump.

Type	Horizontal duplex.
Diameter of steam cylinder, inches	4½
water cylinder, inches	2¾
Stroke of piston cylinder, inches	4

Evaporator Feed Pump.

Type	Horizontal duplex.
Diameter of steam cylinder, inches	2
water cylinder, inches	1½
Stroke of pistons, inches	2¾

There is an evaporator of the Davidson horizontal type, of 2,000 gallons capacity, located on port side in the after end of

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the engine room, with the necessary pumps and pipe connections.

The distiller, of 800 gallons capacity, is of the Davidson vertical type, located at the side of the vessel, above the evaporator.

The air compressor is of the three-cylinder, triple compression type; it is located on the starboard side of the engine room, just aft of the engines.

The steering engine, located in the after part of the engine room, on the center line, is of the horizontal, two-cylinder type, and was built by the contractors from their own designs. It consists of a double-cylinder horizontal engine, driving a worm, which meshes with a worm wheel. The worm wheel turns on a shaft carrying the cable drum, which can be thrown in or out of connection with the worm wheel by means of a hand lever. When thrown out of gear the boat is steered by the hand wheel in the after conning tower. The controlling valve of the engine is actuated by means of a shaft leading to the pilot wheel on the bridge. There is a compensating attachment on the engine which causes it to follow the variations of the movement of the steering wheel very closely, the exact position of the tiller at all times being indicated by the telltale on the bridge.

The electric-light plant is located on the port side of the engine room, near the after bulkhead; it consists of a double-cylinder, vertical, inverted, direct-acting, piston-valve engine, directly connected to a multipolar dynamo; the whole on one base. It was furnished by the General Electric Company. This plant is of 3.6-kilowatt capacity at 800 revolutions per minute. On the port side, behind the dynamo, is the switchboard, on which is mounted the switches controlling the several circuits to different parts of the boat, and also the volt meter, ampère meter, pilot lights, fuse boxes, &c.

SPEED TRIAL.

The torpedo boat *Goldsborough* left wharf number four, Navy Yard, Puget Sound, shortly after noon, March 30, 1905, getting under way for the measured-mile course off Alkao Point at 12·12½ P. M.

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The weather was clear; there was a light southerly breeze blowing, with the sea but slightly disturbed.

The run, of about 12 miles to the course, was made under forced draft, at about 20 knots speed; the upper set of range stakes was passed at 12·47 P. M.

The standardization runs consisted of three pairs of runs over the measured course. The conditions during each double run, as concerns the revolutions, were as uniform as possible, except on the last, when the revolutions dropped from 341.8 to 285.5, owing to a leak in the port feed pipe, necessitating running the evaporator and auxiliary feed pump for the forward boiler.

The time of each run over the course was taken with stop watches, and the revolutions of each engine for the elapsed time were taken from two Pratt & Whitney counters, set up on the deck aft of the engine-room hatch and actuated by means of a cam on the engine shaft below.

The results of the standardizing runs are given in the attached table.

After completing the standardizing runs, during which the speed of 27 knots, as required by the Department's order, was obtained, the hour trial was immediately run off toward Tacoma for one-half hour and return. So many air and steam leaks had developed during the progress of the trial that it was impossible to develop sufficient power to maintain more than an average of 290 revolutions, developing, according to the speed curve, a speed of 25.2 knots.

Hand-picked Cardiff coal was used on these trials and was very skilfully fired, there seldom being any black smoke showing at the funnels, and at no time was any fire observable coming from them; the steam pressure was also well maintained during the runs.

The crew carried on the trials in the engineering department was as follows, from the torpedo boat *Paul Jones:*

Engine Room.—One chief machinist, two first-class machinists, two second-class machinists, four oilers, one electrician, one water tender.

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OVER MEASURED KNOT.

Number of run.	Time.		Knots.	Average revolutions per minute.	Total revolutions.		Average total revolutions.	Steam pressure at engines.	1st receiver.	2d receiver.	Vacuum.	Air pressure.	
					Stbd. engine.	Port engine.						Forward fireroom.	After fireroom.
	m.	s.										inches.	inches.
1st trip, East	2	$50\frac{4}{5}$	21.1	246.0	706	692	699	250	25	0	28	2	4
2d trip, West	2	$52\frac{4}{5}$	20.8	242.0	702	692	697	245	23	0	28	2	4
3d trip, East	2	26	24.7	280.2	691	673	682	255	50	15	28	3	4
4th trip, West	2	$27\frac{1}{5}$	24.5	285.7	706	694	700	250	28	12	28	3	4
5th trip, East	2	0	30.0	*341.8	704	663	683.5	260	80 s. 60 p.	30 s. 21 p.	26	4	5
6th trip, West	2	25	24.8	285.5	702	678	690	215	10	2	20	0	4

* Throttle valves wide open. Cut-off links on all runs, two-tenths down.

1·45 TO 2·45, HOUR RUN. 1·51 TO 2·40 P. M., TIME OF OBSERVATION.

Time.	Knots.	Revolutions per minute.	Total revolutions.		Average total revolutions.	Steam pressure at engines.	1st receiver.	2d receiver.	Vacuum.	Air pressure.	
			Starboard engine.	Port engine.						Forward.	Aft.
2·01	25.8	296	3,048	2,871	2,960	240	100 s. 90 p.	75 20	16	5	4
2·21	...	...	9,021	...	...	Reading on port engine not taken, as escaping steam drowned the signal.					
2·40	25.3	290.3	14,604	13,848	14,226	250	95	26	16	5	4

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Forward Fire Room.—One water tender, one blower-engine tender, two firemen, two coal passers.

Aft Fire Room.—Two water tenders, two blower-engine tenders, four firemen, four coal passers. Three firemen on deck at boiler stops, three men on deck taking data.

The draught and displacement of the boat before and after the trial were as follows, pig iron being used to attain the displacement required.

	Before.	After.
Draught forward, feet and inches	4–10½	4– 4½
aft, feet and inches	9– 1¼	9– 1
mean, feet and inches	6–11⅞	6– 8¾
Displacement, tons	264	247
Mean displacement on trial, tons	255½	
Mean draught on trial, feet and inches	6–10 9/16	

INSPECTION AFTER TRIAL.

Both H.P. piston-rod metallic packings leaked badly.

Port after L.P. top cylinder-head gasket gave way.

Forward pin in the port after rock arm for suspension link of L.P. valve motion snapped off close to the body of the guide block, while on the hour run.

Leak developed in feed pipe to forward boiler while on the sixth run, necessitating starting up evaporator and auxiliary feed pump.

Vapor pipe from evaporator to condenser started to leak on hour run, reducing vacuum from 28 inches to 16 inches.

Top of relief valve on head of the port after low-pressure cylinder kept lifting at intervals during the whole run, wasting a large quantity of fresh water.

All the above steam leaks, together with minor leaks, kept the engine room filled with steam, making it at times almost unendurable, but, notwithstanding this, the attendants kept at their stations and succeeded in keeping all bearings cool.

It is believed that when the boat is placed in commission, with a regular crew attached, who will naturally take interest and keep overhauling and repairing minor defects that will likely occur, the boat will prove to be a very acceptable addition to the service, provided she is not forced to extreme speed, say, not over 26 knots per hour.

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NOTES ON THE PERFORMANCE OF THE THORNYCROFT BOILER IN A MONITOR.

By Lieut. W. T. Cluverius, U. S. N., Member.

To have steamed 28,000 miles since commissioning, three years ago, with nearly one-third of this time consumed in repairing a turret mount at a navy-yard dock, seems a fair test of the performance of the four Thornycroft-Daring boilers of the harbor-defence monitor *Arkansas*.

This type of water-tube boiler has been present in our torpedo craft for several years, and trouble has been found generally in the fact that the upper rows of tubes could not be drowned and corrosion in these tubes readily set in. In the later pattern of these boilers, however, these tubes are flush with the top of the steam drum and are drowned if the boiler is kept entirely full of water. This obtains on the *Arkansas*, and undoubtedly the increased life of a Thornycroft boiler tube is due to this change in design, for on the *Arkansas* not a sign of even slight corrosion has appeared until recently. In fact, no tubes had been lost, except once through careless handling of the feed when newly commissioned, until last month. Two tubes were lost at that time in the vicinity of the steaming level, the only locality in the drum where the tube ends show any evidence of corrosion.

Assuming that the life of a tube depends upon the care which it receives, and that it is not affected by the forcing of the boiler within reasonable limits, the longevity of such tube may be determined through inquiry into the attention given the boiler.

The following is the procedure on the *Arkansas*, and, whether successful or otherwise, it is at least consistent:

When the fires have been burned down and the boilers dis-

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connected, the tubes are swept with a steam lance from top to bottom while there is still pressure showing in the gauge. If in port, the uptake also.

This removes much of the deposit, though not any of that which has settled in the spaces where the nests of tubes contract to enter the steam drum and the wing drum. If neglected, these spaces, for a distance of nearly a foot above the wing drums, and as much outboard from the steam drum, become entirely choked and reduce the heating surface, and therefore the efficiency, of the boiler materially.

To get at this deposit, a tin scraper, together with a stiff whisk broom, is used. Each tube is carefully gone over and the interstices cleaned out. This is necessarily a slow process, but it is time well spent.

Even this does not reach properly the inner corners where the rear nests of tubes enter the wing drums; and if the furnaces are allowed to remain any length of time without cleaning it becomes impossible to clear out these pockets without removing the rear casing of the boiler. As a remedy it is suggested that soot doors be fitted in this casing as in the front casing. Two such doors on each side, one just above the wing drum and the other well up in the casing, would meet the requirements.

As regards reaching the tubes of the middle water drum, much less difficulty has been experienced.

With this attention in port to tubes and uptakes, the boilers have steamed freely for five days without sweeping.

All of the feed water used is distilled on board, no matter whether taken from the sea or the shore. The water is given frequent nitrate tests with a standard solution at the distillers before it enters the reserve tank. The condensers receive the same test. I believe that no salt scale has ever been found in any of the four boilers.

To insure an alkaline state, a small amount of sal soda is put in each main condenser every other day while underway, and in port about a quarter pound once weekly.

To prevent the passage of grease into the boilers, no oil is

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allowed in the cylinders and but little for swabbing the rods. The dynamo engines being steepled, with low-pressure cylinder below, and running in oil as they do, some oil is drawn up the low-pressure piston rod and finds its way to the feed and filter tank.

The water in this tank is constrained to pass through three thick layers of loofa—a most satisfactory filtering agent—before it reaches the feed pumps. This loofa is removed every fortnight and given a soda bath, being entirely cleansed of oil. The life of a piece of loofa is about two months. It is also used in the distiller filters with excellent results.

Once each month the boiler drums are opened, cleared of zinc baskets, baffles, etc., thoroughly wiped out and inspected. A solution of 25 pounds of sal soda is then placed in the boiler; it is closed up, filled to steaming level and fires are started. It is boiled out for six hours at a pressure of 50 pounds. The boiler is now blown down, opened and again wiped out. The drum fittings are overhauled, after which the boiler is closed and filled with water.

Judging by the present condition of the visible portions of the tubes—and it is impossible to determine the condition of a bent tube throughout its length—coupled with the uniform attention which the *Arkansas* boilers have received since installment, I would submit that the period of *complete* efficiency of a Thornycroft boiler in a vessel of the monitor type, performing the same service as the *Arkansas*, is three years. After that, watch the tubes closely.

There are two furnaces to each boiler, and the small grate area of each (24.8 square feet) necessitates a more frequent cleaning of fires than ordinarily. In free route at sea, with the average quality of coal and under natural draft, it is often found desirable to clean one furnace of each boiler or half of the whole number of furnaces every watch. With clean coal, free from clinker or under forced draft, one quarter of all the furnaces suffices.

The use of the blowers, so often demanded while steaming in squadron, through foul condition of hull or with poor fuel,

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has proved very trying to the furnace linings. Not only do the walls require constant patching, but the casings themselves suffer eventually, and the least warping of the front casings produces leaky furnace-door frames and also endangers the front-lining brickwork.

In common with all boilers of this type, the proportion of heating to grate surface is large (47.3 to 1), and the gases well baffled. The *Arkansas* can steam, under normal conditions of hull and sea, 2,700 miles at a ten-knots speed.

Probably the most interesting comparison in the operation of these boilers is that of their performance with a clean hull and with a foul one, other conditions being equal.

A, is a sixty-hour trial during a run to the West Indies, clean hull, natural draft.

B, is a sixty-one-hour trial on the run North, very foul hull, natural draft.

	A.	B.
Distance run, knots	621.6	597.2
Average speed, knots	10.36	9.8
revolutions	150.1	156.4
Revolutions per knot	14.5	16.
Coal per day, this rate, tons	31.	41.
Knots per ton of coal	8.	5.7
Steaming radius, miles	2,672.	1,903.8

This shows a 25 per cent. increase in the daily coal consumption due to foulness alone; so that this factor, together with a heavy head sea, most of which is shipped, becomes a serious question in a monitor.

Following is a table which may also be of interest:

—	A.	B.	C.	D.
Revolutions per minute	206.5	197.1	180.1	186.7
Speed per hour, knots	13.5	12.7	10.7	11.6
Total horsepower developed	*2,500	1,826.6	2,080.4	2,218.9
Pounds of coal per hour	7,200	6,720	5,471	6,859.5
sq. ft. grate surface	36.4	33.9	27.5	34.6
I.H.P. per hour	2.9	3.6	2.6	3.1

*Approximated.

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A, is the best hour's run the ship has made, hull clean.
B, is the two-hours official acceptance trial, hull clean.
C, a three-hours monthly speed trial, hull foul.
D, a four-hours monthly speed trial, hull foul.
All under nearly similar conditions of forced draft.

Since the three harbor-defence monitors in the North Atlantic Fleet have spent but little time defending harbors in these days of peace and are cruising in squadron with no reserve speed left them, the items of clean hulls and clean boilers, good coal and good firing, are very essential items indeed.

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WEIGHTS OF MACHINERY.

By W. F. Sicard, Associate.

The subject of machinery weights is one that should be considered carefully by all shipbuilders, particularly in connection with naval work, where an excess over the contract allowance involves a penalty. It is safe to say that nearly all the vessels of the new Navy have averaged somewhat over the contract weight and have been penalized in consequence. In isolated cases it might be somewhat difficult to tell whether such excess weights were unavoidable or not, but when the weights of a considerable number of vessels of the same class are available for comparison a reasonable conclusion can be reached as to what the weights should have been for the average ship of such class. The Navy has now enough vessels completed to make such comparisons of value, and it is to the interest of both the Government and the shipbuilders if it can be made convincingly clear that such overweights as have resulted could have been avoided without imperiling the strength or durability of the machinery and without restricting, within too narrow limits, that natural variety in design which is the stepping stone of engineering progress.

A careful consideration of the returned weights of naval vessels convinces the writer that most of the instances in which the machinery installations have been overweight, up to this time, could have been avoided, and it is hoped that the following pages will make clear, in detail, where weight might have been saved.

There is, perhaps, no subject connected with the design of marine machinery concerning which more distrust and uncertainty is felt than the preliminary estimate of weights. That this should be so with vessels of unusual or untried types is not surprising; but when, as is usually the case, the

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machinery is of an established type and one that has been installed in a number of vessels, the weights of which have been accurately recorded, there is no reason why, for a new vessel of similar design, a careful estimate should not approximate very closely to the finished weight.

Early in the course of design of every vessel there must be a definite and accurate assignment of weights to each of the important elements, such as armor, armament, machinery, equipment, hull, etc. Every design is a compromise in which these elements, many of them conflicting and all seeming to demand an undue share of the available weight, have been critically compared and harmonized until, when the design is finally settled, they bear a definite relation to each other, this relation being dependent upon the type of vessel and upon the service she will be required to perform. In the relation of the final design each element has its definite allotment of weight that cannot be changed except at the expense of some other element, for any variation in the weight of one element affects both the mutual relation of the others and their total weight, and the success of this part of a design depends largely upon an accurate preliminary approximation to what the finished weights will be.

In nearly all vessels of war the most important elements or characteristics are those of offence and defence, and on a given displacement all weight added to any of the other elements above their proper apportionment of the remaining available weight must be gained by reducing either or both of these vital qualities, therefore reducing the military efficiency of the vessel, and consequently, more or less, her reason for existence.

A consideration of the whole subject of the weight of all the different elements of a design and their mutual relation is much too broad to be discussed, in all its varying phases, within the limits of this paper, and therefore only what is commonly included in the term machinery weights will be considered here.

This question of machinery weight is considered so import-

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ant by the Navy Department that a penalty is required for every ton in excess of the weight fixed by the contract.

The clause referring to this subject on the latest form of contract for large naval vessels of the first class reads as follows:

"* * * and that the total weight of said machinery, including engines, boilers and appurtenances, all fixtures in engine and firerooms, smoke pipes, distilling apparatus, spare parts, heating apparatus, tools in workshop, and water in boilers, condensers, pumps, pipes and stern tubes, that is to say, all weights under cognizance of the Bureau of Steam Engineering, except supplies furnished by the Government (but not including the reserve feed water in double bottom or tanks or spare parts of the machinery not usually carried on board, as enumerated in the specifications), shall not exceed * * * tons, this weight to be determined from the certified records of the actual weight of the parts of the machinery as they are sent on board the vessel to be connected up, except the weight of the contained water, which shall be calculated from the actual volumes in steaming condition, as shown on the certified drawings of the completed machinery; the weight to be calculated for salt water, except for those parts where fresh water only is used; that if said total weight be exceeded, a deduction of five hundred dollars ($500) a ton shall be made from the contract price of the vessel for each ton of weight in excess of that stipulated, and that if said total weight be exceeded by five (5) per cent., a further deduction of ten thousand dollars ($10,000) shall be made from the price of the vessel * * *."

In order to fulfill this requirement, after a contract is entered into and the work is begun every object under the cognizance of the Bureau of Steam Engineering is weighed by the contractor in the presence of a naval inspector of machinery, or one of his assistants, before being placed on board the vessel. The weights are carefully recorded by the inspector on special blank forms supplied for the purpose, and when the vessel is finished these forms are forwarded to the Bureau,

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where the weights and centers of gravity are computed, and, in case there is over weight, the amount of penalty is determined in accordance with the stipulations of the contract as quoted above.

These forms, called the finished-weight sheets, are bound and kept on file at the Bureau for reference and for use in basing estimates of weights and centers of gravity of new machinery of a similar type.

Of course, all the separate items that appear on the weight sheets for naval vessels are arranged according to the Bureau's classification of weights. It often happens, however, say in the case of preliminary or unofficial estimates for new work from shipbuilders, or in statements of weight of merchant work, that the weights are given according to quite different forms of grouping and classification.

As nearly every shipbuilding firm has its own system of keeping weight returns, and as these systems vary somewhat in respect to the division of different items into groups, as well as in regard to the different subjects included under machinery weights, it is often very difficult to form comparisons, or a clear idea as to exactly what is included, when the weights of machinery installations are given by different firms in their own classification.

It would be a great advantage all around, and a step in the direction of uniformity, if outside firms would adopt the Bureau's classification of weights for all marine work. It may be urged that this cannot be conveniently done, owing to the fact that the division of work employed by the Navy Department under its bureau system does not obtain in private yards. This objection, however, can easily be overcome by adding new groups, containing the items where the jurisdiction differs, this being simpler, because the shipbuilding firms almost invariably include in their machinery weights items which, in Department work, are divided among several bureaus. Therefore, to make the change without affecting their own practice, it would only be necessary to make a rearrangement of their existing forms of classification to the extent of taking out of

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their machinery weights subjects not included by the Bureau of Steam Engineering, and forming these subjects into certain new groups which, when used in naval work, could be readily credited to the bureaus concerned, and when used in their own work could be placed where desired.

Aside from the inconvenience of having different systems, it certainly involves more work and expense if, after the objects are weighed and classified in accordance with the bureau system, they must be reclassified according to another, which, by the way, is usually not as good nor as well adapted for estimate of weight or cost.

By having a definite system of weight classification for *all* machinery weights it will be found that accurate estimates and comparisons can be much more quickly made and actual weights will be much more available for use.

The system of classifying and keeping weights adopted by the Bureau of Steam Engineering has been in use for some years and has proved accurate and complete. Briefly it is as follows:

All objects under the Bureau's cognizance are divided into a number of subjects. Similar subjects or those closely related are collected into groups, the weight of each group, for each division of the ship, being kept on separate weight sheets. For the purpose of placing these groups in their proper divisions or compartments the weights are divided into—

Bow weights, including all weights of machinery forward of the forward bulkhead of main machinery compartments.

Stern weights, including all weights abaft the after bulkhead of main machinery compartments.

The weights between the forward and after bulkheads of main machinery compartments are divided according to the actual transverse bulkheads in the ship and named according to location, "*After*," "*Middle*," or "*Forward*," "*Fireroom Weights*;" "*Forward*" or "*After*" "*Engine-room Weights*." Should there be bunkers or auxiliary machinery rooms in this space (separated from it by transverse bulkheads) an appropriate title for the space so included is assumed.

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By "main machinery compartments" is meant all space included between the forward and after bulkheads inclosing the main engines and boilers. All bulkheads are supposed to extend upward indefinitely and to each side of the ship.

The individual subject division of all the machinery weights and the arrangement of the same into groups, according to the Bureau's classification, is as follows:

GROUP I.—MAIN ENGINE CYLINDERS.

Subject.

1. H.P. cylinders, including cylinder casings, liners, covers; valve chests, liners and covers; bolts and nuts.
2. 1st I.P. cylinders, including same.
3. 2d I.P. cylinders, including same.
4. L.P. cylinders, including same.
5. Relief and drain valves and receiver safety valves for main-engine cylinders.
6. Main receiver piping.
7. Main piston rod and valve-stem stuffing boxes and packing.

GROUP II.—SHAFTING.

Subject.

1. Main crank shafts.
2. Main line shafts.
3. Main propeller and stern-tube shafts, with nut and keys.
4. Main thrust shafts.
5. Main shaft couplings (special), friction wheels, brakes and clutch couplings.
6. Main shaft coupling bolts and nuts.

GROUP III.—MAIN ENGINE FRAMING AND BEARINGS.

Subject.

1. Bedplates, including crank-shaft brasses, caps and bolts, and foundation bolts and nuts.
2. Columns and tie rods and housings, with bolts and nuts.
3. Crosshead guides, with bolts and nuts.
4. Thrust bearings, with bolts and nuts.
5. Line-shaft bearings, with bolts and nuts.
6. Stern tube, stern-bracket and strut bearings, with bolts and nuts. (Not including tube, bracket or strut.)
7. Bulkhead stuffing boxes around shafting (complete).

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GROUP IV.—RECIPROCATING PARTS OF MAIN ENGINES.

Subject.

1. H.P. pistons, complete.
2. 1st I.P. pistons, complete.
3. 2d I.P. pistons, complete.
4. L.P. pistons, complete.
5. Piston rods, with nuts, keys, etc.
6. Crossheads complete, including slippers.
7. Connecting rods; rods, brasses, straps, bolts and nuts, complete.

GROUP V.—MAIN ENGINE VALVE GEAR.

Subject.

1. Eccentrics.
2. Eccentric straps and bolts.
3. Eccentric rods, brasses and bolts.
4. Links and blocks, complete.
5. H.P. valves, complete.
6. 1st I.P. valves, complete.
7. 2d I.P. valves, complete.
8. L.P. valves, complete.
9. Valve stems, brasses, nuts, keys and balance pistons.
10. Valve-stem crossheads.
11. Valve rock shafts.
12. Suspension links, crosshead guides, radius rods and other gear.
13. Reversing engines and gear, complete, with shafts, brackets and arms.

GROUP VI.—MAIN CONDENSERS.

Subject.

1. Main condensers, shells, heads, tubes, etc., complete.

GROUP VII.—MAIN AIR AND CIRCULATING PUMPS.

Subject.

1. Main air pumps, barrel, liner, covers, bucket, rods, etc., complete to piping.
2. Main air-pump engines, complete to piping.
3. Circulating pump, complete to piping.
4. Circulating-pump engines, complete to piping.

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GROUP VIII.—PROPELLERS.

Subject.

1. Main propellers, without shaft nut, but with cap and all other bolts and nuts.

GROUP IX.—BOILERS.

Subject.

1. Main boilers, including shells, butt straps, rivets, heads, furnaces, combustion chambers, tubes, tube sheets, and all rivets and braces, and manhole plates, and, in the case of water-tube boilers, including Subject 6, Group XIV.
2. Auxiliary boilers, including same.

GROUP X.—BOILER FITTINGS.

(*a*) **Main boilers.** (*b*) **Auxiliary boilers.**

Subject.

1. Grate bars and bearers.
2. Bridge walls and fittings.
3. Furnace fronts, doors, ash-pit doors, etc., lazy bars.
4. Ash pans.
5. Other furnace fittings.
6. Dry pipes, and all other internal pipes, circulating apparatus hangers, etc.
7. Zincs, hangers, baskets and bolts.
8. Water gauges and cocks; safety, sentinel and air valves; drain, air, soda and other cocks attached to boilers.
9. Boiler fastenings for securing boilers in ship (not including saddles).

GROUP XI.—SMOKE PIPES AND UPTAKES.

(*a*) **Main boilers.** (*b*) **Auxiliary boilers.**

Subject.

1. Uptakes, including all weights from boiler to base of smoke pipe.
2. Smoke pipes, including all weights of pipes, guys, etc.

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GROUP XII.—STEAM AND EXHAUST PIPES AND VALVES.

Subject.

1. Main steam pipes and valves, including pipe, flanges, bolts and nuts, hangers and fittings.
2. Auxiliary steam pipes and valves, including same.
3. Branch steam pipes to machinery under Bureau of Steam Engineering, including same.
4. Main exhaust piping.
5. Auxiliary exhaust piping.
6. Branches of exhaust piping to machinery under Bureau of Steam Engineering.
7. Escape pipes, whistles, mufflers, etc.

GROUP XIII.—SUCTION AND DISCHARGE PIPES AND VALVES.

Subject.

1. Main injection (bilge and sea) pipes, valves and fittings.
2. Main outboard delivery pipes, valves and fittings.
3. Main air-pump suction and discharge pipes, valves and fittings.
4. Main and auxiliary feed-pump pipes, valves and fittings.
5. Other auxiliary pump pipes, valves and fittings (except water-service pipes).
6. Pump relief valves.
7. Blowpipes and valves outside of boilers, and connections for pumping out boilers.
8. Drain pipes and traps (to machinery under Bureau of Steam Engineering).

NOTE.—Branches from pumps to fire main and main drain pipes will be taken up to flanges on those pipes. All suction pipes to bilges of machinery compartments will be included.

GROUP XIV.—LAGGING AND CLOTHING.

Subject.

1. Lagging and clothing of main cylinders.
2. Lagging and clothing of cylinders of auxiliary engines, feed heaters, etc.
3. Lagging and clothing of main condensers.
4. Lagging and clothing of auxiliary condensers.
5. Lagging and clothing of feed tanks.
6. Lagging and clothing of main boilers.
7. Lagging and clothing of auxiliary boilers.
8. Lagging and clothing of pipes and valves.
9. Lagging and clothing of dynamo condensers and pumps.

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GROUP XV.—FLOORING, GRATINGS, ETC.

Subject.

1. Floor plates, supports, bolts, etc.
2. Gratings, platforms, supports, bolts, etc.
3. Ladders, supports, bolts, etc.
4. Hand rails, supports, bolts, etc.
5. Guards, supports, bolts, etc.

GROUP XVI.—AUXILIARIES.

Subject.

1. Feed pumps for main boilers.
2. Feed pumps for auxiliary boilers.
3. Fire and bilge pumps.
4. Water-service pumps.
5. Other auxiliary steam pumps (not including evaporator pumps).
6. Auxiliary condenser with air and circulating pumps.
7. Shaft pumps.
8. Turning engines and gear.
9. Ash hoists.
10. Blowers and engines (not including ventilating blowers).
11. Coal winches.
12. Other auxiliary engines under Bureau of Steam Engineering.
13. Dynamo condensers, with air and circulating pumps.

GROUP XVII.—FITTINGS AND GEAR.

Subject.

1. Feed and filter tanks, fittings, and vapor pipes.
2. Oil, tallow, soda and waste tanks and fittings.
3. Water-service pipes, main and auxiliary machinery.
4. Lubricating gear, main and auxiliary machinery.
5. Lifting gear for handling parts of machinery.
6. Gear for working valves from decks.
7. Gauges, thermometers, clocks, counters, telegraphs, etc.
8. Separators.
9. Gear for operating engines.
10. Counter gear.
11. Indicator gear.
12. Governors.
13. Air ducts (not including ventilating ducts).
14. Feed-water heaters, grease extractors, and other fittings.
15. Other gear.

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GROUP XVIII.—WATER.

Subject.

1. Water in condensers. (Taken full on salt-water side only.)
2. Water in feed tanks. (Taken two-thirds full.)
3. Water in stern tubes.
4. Water in main boilers. } Taken 4 inches above highest heating service for Scotch boilers and at working level of other boilers.
5. Water in auxiliary boilers. } Taken 4 inches above highest heating service for Scotch boilers and at working level of other boilers.
6. Water in all auxiliary pumps, pipes and auxiliary machinery necessarily in use when running engines at full power.

GROUP XIX.—STORES, TOOLS AND SPARE PARTS.

Subject.

1. Tools in workshop, including engine or motor.
2. Other tools, with racks, etc.
3. Spare parts, with bolts, etc., for securing.
4. Stores supplied by contractors.
5. Hose and racks.
6. Indicators and instruments not always in use.

GROUP XIXa.—GOVERNMENT STORES.

Subject.

1. Stores supplied by Government, under titles B and Y.

GROUP XX.—MISCELLANEOUS MACHINERY, ETC.

Subject.

1. Distilling plant, including distillers, evaporators, pumps, pipes, lagging and clothing, fittings and water.
2. Heating plant, including radiators, pipes, valves, lagging and clothing, and fittings.
3. Hydraulic pumping plant, including pumps, accumulators and all fittings, and water.
4. Steam turret-turning plant.
5. Refrigerating plant, including ice machine and all refrigerating pipes.
6. Air-compressor plant, under Steam Engineering.

GROUP XXI.—OTHER MISCELLANEOUS MACHINERY, ETC.

This group is composed of certain subjects which are not under the Bureau of Steam Engineering, and its weights are not included.

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GROUP XXII.—CONNECTIONS UNDER STEAM ENGINEERING TO "OTHER MISCELLANEOUS MACHINERY."

Subject.

1. Branch steam pipes and valves.
2. Branch exhaust pipes and valves.
3. Suction and discharge pipes and valves.
4. Drain pipes, traps and valves.
5. Lagging and clothing.
6. Water.
7. Lubricating gear and water service.

At first glance this system of classification may appear to be in unnecessary detail, but, while seemingly elaborate, it is in reality very simple and flexible, permitting the finished weights to be so recorded that the actual weight of even a small object or the part of a subject can be quickly found if desired, thus being of use in estimating for cost of new parts. When used for estimate it offers several different methods for computing the weights. It permits either a rough and general estimate to be quickly made, based on the total weight, or an estimate based on per cent. of compartment weights in similar vessels, or a group estimate, or a subject estimate, this latter being capable of great elaboration and refinement in detail.

It is often useful to know the machinery weight in any compartment or its per cent. of the total weight, and if this division of weight is made at the time of estimating, it really involves very little extra work, besides permitting a much closer estimate to be made.

It is the writer's opinion that there are no formulae of any use for estimating machinery weights. A preliminary estimate of the total weight necessary for the machinery installation of a new vessel is usually required before the size of various objects and their location is definitely determined. This preliminary estimate must often be made when possibly only the horsepower, the general type of engines and boilers and a few very indefinite general particulars are known, and there is usually at this stage of the work no drawing for locat-

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ing the machinery and getting the length of shafting and such items that very much affect the weight.

Under such conditions it is manifestly impossible to make any kind of a detailed estimate, and the best way to figure this first and preliminary weight is to base it on pounds per indicated horsepower, selecting a constant from a table of weights of machinery having similar salient features. This gives a more accurate result than might be expected for what might be minor inaccuracies in group and subject were a detailed estimate made, tend to average, and, in dealing with the larger sum, they disappear and are lost. Later, when the type and size of the machinery is more definitely settled, and the drawings are far enough advanced to scale for size of parts and location of centers of gravity, another estimate should always be made. This time it is well to estimate on the subjects in detail, according to the classification, bearing in mind that the general tendency is always to underestimate, and checking the new weights by subject, group and compartment comparison with other similar vessels, and checking their total with the first estimate. This detailed estimate, with its cross checks and the facility it offers for comparison, should show at once where, if at all, an error has been made, and should show to what particular group, or item, or subject, an excess or deficiency is due.

This estimate, based on individual subjects, does not take as long to make as might be supposed. A glance at the accompanying finished-weight sheet, which shows group IV and all its subjects, will make it apparent how closely the weight of a new engine of a similar design could be figured from such detail as this sheet shows.

The method of keeping weights adopted by the Bureau of Steam Engineering is as follows:

The actual weight of each subject in detail, and the distances of its center of gravity from fixed vertical and horizontal planes, are entered on the following blank forms, called the finished-weight sheets.

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ACTUAL MACHINERY WEIGHTS, U. S. S. * * * * *.

For estimate, weight per cubic inch:
Cast iron = 0.261 lbs. Wrought steel = 0.283 lbs.
Wrought iron = 0.28 lbs. Composition = 0.316 lbs.
Cast steel = 0 283 lbs. Water, to suit temperature.

Subject.	Description of part.	Weight in pounds.	Center of gravity (aft = +, forward = —). Horizontal from frame 60.	Vertical above base line.
1.	H.P. piston, complete:			
	S. Piston, 9 inches diameter	46		
	4 bolts and nuts, $\frac{3}{4} \times 2\frac{3}{4}$ inches	3		
	Piston ring	2.5		
	P. Piston, details as above	52.5		
1.	H.P. pistons, total	104	+ 17.38	11.34
2.	I.P. piston, complete:			
	S. Piston, $15\frac{1}{4}$ inches diameter	86.5		
	6 bolts and nuts, $\frac{3}{4} \times 2\frac{3}{4}$ inches	4.5		
	Piston ring	5		
	P. piston, details as above	96.5		
2.	I.P. pistons, total	192.5	+ 20.29	11.32
4.	L.P. piston, complete:			
	S. Piston, $25\frac{1}{4}$ inches diameter	190		
	8 bolts and nuts, $\frac{3}{4} \times 2\frac{3}{4}$ inches	6		
	Piston ring	10		
	P. piston, details as above	206		
4.	L.P. pistons, total	412	+ 22.70	12.63
5.	Piston rods with nuts and keys:			
	S. 3 main piston rods, 3 ft. $6\frac{25}{32}$ in. over all	187.5		
	3 nuts, $1\frac{5}{8}$ inches diameter	8		
	P. 3 main piston rods and nuts	195.5		
5.	Piston rods with nuts and keys, total	391	+ 19.94	9.76
6.	Crossheads and slippers, complete:			
	S. 3 crosshead brasses, $3\frac{3}{8}$ in. dia. $\times \frac{5}{8}$ in.	87		
	6 bolts and nuts, $1\frac{3}{8} \times 9\frac{9}{16}$ inches	33		
	3 crosshead slippers, $8 \times 8\frac{3}{4}$ inches surface	96		
	12 tap bolts, $\frac{3}{4} \times 1\frac{1}{2}$ inches	4.5		
	P. 3 crosshead brasses and slippers, details as above	220.5		
6.	Crossheads and slippers, total	441	+ 19.86	9.06
7.	Connecting rods, complete:			
	S. 3 connecting rods, 4 ft. $4\frac{1}{2}$ in. bet. centers	373		
	3 connecting-rod brasses	294		
	6 bolts and nuts, $1\frac{1}{2} \times 14$ inches	54		
	P. 3 connecting rods, details as above	721		
7.	Connecting rods, brasses, etc., total	1,442	+ 19.62	7.48

When the vessel is completed, the finished-weight sheets are forwarded to the Bureau, where the weights in detail, with subjects properly collected into groups, are entered on the

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following blank form, and the moments computed for each subject of each group, from which are obtained the centers of gravity of each group.

ACTUAL MACHINERY WEIGHTS, U. S. S. * * * * *.

Group.	Description of part.	Total weights.		Horizontal.		Vertical.	
		Pounds.	Tons.	From frame 78.5.	Moments.	Above base line.	Moments.
III.	Bedplates	143,621	64.12	28.5	1,827.42	7.5	480.9
	Columns and tie rods	76,329	34.08	27.8	947.42	13.7	466.9
	Crosshead guides	4,779.5	2.13	21.6	46.01	16.1	34.29
	Crosshead guides	4,684.5	2.09	21.6	54.14	16.1	33.65
	Crosshead guides	4,614.5	2.06	32.9	67.77	16.2	33.37
	Crosshead guides	4,834.5	2.16	32.9	71.06	16.2	34.99
	Crosshead guides	3,619.5	1.61	15.9	25.60	16	25.76
	Crosshead guides	3,627.5	1.62	15.9	25.76	16	25.92
	Crosshead guides	3,544.5	1.58	39.6	62.57	16.3	25.75
	Crosshead guides	3,547.5	1.58	39 6	62.57	16.3	25.75
	Thrust bearings	35,313.2	15.77	52	820.04	8	126.16
	Total	288,516	128.8	31.14	4,010.36	10.2	1,313.44
IV.	H.P. piston	3,628	1.62	21.5	34.83	20.5	40.5
	I.P. piston	5,534.5	2.47	32.8	81.02	22.9	56.56
	L.P. piston	3,637	1.62	15.8	25.60	20.8	33.7
	L.P. piston	3,577	1.6	15.8	25.28	20 8	33.28
	L.P. piston	3,647	1.63	39.5	64.39	22.9	37.33
	L.P. piston	3,637	1.62	39.5	63.99	22 9	37.1
	H.P. piston rod	1,634	.73	21.6	15.77	21.8	15.91
	Piston rods	1,639	.73	21.6	15.77	21.8	15.91
	Piston rods	1,606	.72	32.9	23.69	19.6	14.11
	Piston rods	1,606	.72	32 9	23.69	19.6	14.11
	Piston rods	1,557	.69	15.9	10 97	17.6	12.14
	Piston rods	1,517	.68	15.9	10.81	17.6	11.97
	Piston rods	1,532	.68	39.6	26.93	19.6	13.33
	Piston rods	1,547	.69	39.6	27.32	19.6	13.52
	Crossheads	2,436.5	1.09	21.6	23.54	17.9	19.51
	Crossheads	2,440 5	1 09	21.6	23.54	17.9	19.51
	Crossheads	2,438.5	1.09	32.9	35.86	15.8	17.22
	Crossheads	2,447.5	1.09	32.9	35.86	15.8	17.22
	Crossheads	2,237.5	1	15.9	15.9	13.8	13.8
	Crossheads	2,238.5	1	15.9	15.9	13.8	13.8
	Crossheads	2,245.5	1	39.6	39.6	15.9	15.9
	Crossheads	2,238.5	1	39.6	39.6	15.9	15.9
	Connecting rod	7,234	3.23	21.7	70.09	13	41.99
	Connecting rod	7,209	3 22	21.7	69.87	13	41.86
	Connecting rod	7,214	3.22	33	106.26	11	35.42
	Connecting rod	7,189	3.21	33	105.93	11	35.31
	Connecting rod	7,209	3.22	16	51.52	8.9	28.66
	Connecting rod	7,139	3 19	16	51.04	8.9	28.39
	Connecting rod	7,179	3.20	39.7	127.04	11.1	35.52
	Connecting rod	7,144	3.19	39 7	126.64	11.1	35.41
	Total	112,538.5	50.24	27.63	1,388.25	15.22	764.89
V.	Eccentrics	9,979	4.46	27.6	123.1	8.3	37.02
	Eccentric straps	13,493	6.02	27.6	166.15	8.4	50.57
	Eccentric rod	382	.17	25.7	4.37	13.8	2 35
	Eccentric rod	384	.17	25.7	4.37	13.8	2.35
	Eccentric rod	402	.18	26.1	4.7	13.8	2.48
	Eccentric rod	405	.18	26.1	4 7	13.8	2.48
	Eccentric rod	468	.21	28.1	5.9	14.3	3
	Eccentric rod	471	.21	28.1	5.9	14.3	3
	Eccentric rod	469	.21	10.7	2.25	14	2.94
	Eccentric rod	469	.21	10.7	2.25	14	2.94
	Eccentric rod	469	.21	44.8	9.41	13.7	2.88
	Eccentric rod	470	.21	44.8	9.41	13.7	2.88
	Eccentric rod	494	.22	28.5	6.27	13.4	2.95
	Eccentric rod	496	.22	28.5	6.27	13.4	2 95
	Eccentric rod	500	.22	11.1	2.44	13.9	3 06
	Eccentric rod	496	.22	11.1	2.44	13.9	3 06
	Eccentric rod	495	.22	44.4	9.77	14.1	3.1
	Eccentric rod	497	.22	44.4	9.77	14.1	3.1
	Links and blocks	427	.19	25.7	4.88	16.5	3 14

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All the groups are then combined on sheets similar to the following, and the centers of gravity computed for the sums of all the groups in each compartment.

ACTUAL MACHINERY WEIGHTS, U. S. S. * * * * *.

Group.	Compartment.	Total weight, in tons.	Horizontal moments: foot-tons. Forward of frame No. 78½.	Vertical moments: foot-tons. Above base line.
	Forward boiler room.			
IX.	Boilers	176.02	22,297.99	2,411.48
X.	Boiler fittings	64.43	8,153.44	628.68
XI.	Smoke pipes and uptakes	46.27	6,118.56	1,829.7
XII.	Steam and exhaust pipes and valves	14.76	1,874.71	358.24
XIII.	Suction and discharge pipes and valves.	13.57	1,683.59	142.32
XIV.	Lagging and clothing	1.79	227.49	35.
XV.	Flooring, gratings, etc	11.27	1,464.07	82.63
XVI.	Auxiliaries	11.82	1,496.68	277.8
XVII.	Fittings and gear	1.45	186.14	32.7
XIX.	Stores, tools, spares	13 49	1,826.02	169.54
XX.	Miscellaneous machinery	2.68	347.06	91.3
XXII.	Pipes connecting to XXI	.94	142.86	21.25
	Total without water	358.49	45,818.61	6,080.64
XVIII.	Water	42.59	5,405.95	594.92
	Total with water	401.08	51,224.56	6,675.56
	Middle boiler room.			
IX.	Boilers	117.50	9,987.11	1,609.75
X.	Boiler fittings	42.95	3,650.87	419.06
XI.	Smoke pipes and uptakes	69.1	5,879.45	3,163.3
XII.	Steam and exhaust pipes and valves	12.	1,019.55	276.82
XIII.	Suction and discharge pipes and valves.	10.89	929.50	121.04
XIV.	Lagging and clothing	1.23	104.26	24.46
XV.	Flooring, gratings, etc	7.41	629.65	56.07
XVI.	Auxiliaries	9.01	760.4	185.6
XVII.	Fittings and gear	.96	81.39	22.48
XIX.	Stores, tools, spares	6.43	850.12	136.72
XX.	Miscellaneous machinery	1.48	122.68	51.26
	Total without water	278.96	24,014.98	6,066.56
XVIII.	Water	28.49	2,419.87	398.44
	Total with water	307.45	26,434.85	6,465.

Finally, the weights and moments of all the compartments are collected on a "summary" sheet, and the center of gravity of the entire machinery installation computed.

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ACTUAL MACHINERY WEIGHTS, U. S. S. * * * * *.

Compartment.	Total weight.	Horizontal moments: foot-tons.			Vertical moments: foot-tons.	
		Forward of frame No. 78½.		Abaft frame No. 78½.	Above base line.	
	Tons	*Distances*	*Moments*	*Moments*	*Distances*	*Moments*
Summary without water:						
Stern	139.1	...	...	16,650.72	...	1,463.5
Both engine rooms	677.38	...	...	18,128.65	...	10,832.62
Bunker	50.92	...	465.23	...	...	1,018.26
After boiler room	359.76	...	15,663.11	...	...	6,149 06
Middle boiler room	278.96	...	24,014.98	...	...	6,066.56
Forward boiler room	358.49	...	45,818.61	...	...	6,080.64
Bow	33.44	...	5,982.74	...	...	821.73
	...	...	91,944.67	...	...	...
	...	...	34,779.37	34,779.37	...	...
Total without water	1,898.05	30 12	57,165.30	...	17.09	32,432.37
Summary with water:						
Stern	141.25	...	...	16,869.38	...	1,482.42
Both engine rooms	733.50	...	...	19,599.87	...	11,551.82
Bunker	51.21	...	467.55	...	...	1,022.04
After boiler room	402.73	...	17,507.35	...	...	6,750.03
Middle boiler room	307.45	...	26,434.85	...	...	6,465.
Forward boiler room	401.08	...	51,224.56	...	...	6,675.56
Bow	34.08	...	6,082.99	...	...	826.47
	...	...	101,717.3	...	...	...
	...	...	36,469.25	36,469.25	...	...
Total with water	2,071.3	31.5	65,248.05	...	16.79	34,773.34
XIXa.—Government stores	40.	...	...	...	...	...
Total machinery weight	2,111.3	...	...	...	...	...

We have now been working gradually upward, first with subjects, then groups, then compartments, until the total for the ship is reached. As the centers of gravity are rarely used, in comparison with the weights, the following sheet has been evolved, and it is the most useful and convenient method of stating the weights, showing, as it does completely, both the weights of all the groups and their relative location in the vessel.

For the purpose of comparing the weights of marine machinery installation, a number of cases have been selected in which different types of vessels having machinery of different powers are employed. In all cases the vessels compared are sister ships, and those built by different firms are selected, so as to show in what way and to what extent the machinery weights are affected by the differences that exist in engineering practice. In all of the following cases, except the three ships of the *Maine* class, the builders have developed their

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ACTUAL MACHINERY WEIGHTS OF U. S. S. * * * * *.

Groups.	Stern.	After engine room.	After bunker.	After fireroom.	After middle fireroom.	Forward fire-room.	Bow.	Not placed.	Total.
I.—Main-engine cylinders	...	175.03	...	...	...	...	...	...	175.03
II.—Shafting	78.54	44.16	...	...	...	...	...	...	122.7
III.—Main-engine framing and bearings	6.43	128.8	...	...	...	...	...	...	135.23
IV.—Reciprocating parts of main engines	...	50.24	...	...	...	...	...	...	50.24
V.—Main-engine valve gear	...	38.2	...	...	...	...	...	...	38.2
VI.—Main condensers	...	47.55	...	...	...	...	...	...	47.55
VII.—Main air and circulating pumps	...	31.29	...	...	...	...	...	...	31.29
VIII.—Propellers	28.23	...	...	...	...	...	...	...	28.23
IX.—Boilers	...	...	...	176.08	117.5	176.02	...	...	469.6
X.—Boiler fittings	...	...	...	64.44	42.95	64.43	...	...	171.82
XI.—Smoke pipes and uptakes	...	...	...	47.44	69.1	46.27	...	...	162.81
XII.—Steam and exhaust pipes and valves	...	23.41	4.04	18.95	12.	14.77	1.05	...	74.22
XIII.—Suction and discharge pipes and valves	.15	36.3	1.47	15.27	10.89	13.57	.97	...	78.62
XIV.—Lagging and clothing	...	9.11	.38	1.99	1.24	1.79	.31	...	14.82
XV.—Flooring, gratings, etc	1.81	17.56	.92	11.86	7.41	11.26	...	...	50.82
XVI.—Auxiliaries	...	25.97	...	11.81	9.01	11.82	4.69	...	63.3
XVII.—Fittings and gear	.31	35.66	.21	1.45	.97	1.44	1.21	...	41.25
XIX.—Stores, tools, spares	13.86	8.21	16.15	7.92	6.42	13.49	.36	...	66.41
XX.—Miscellaneous machinery	7.16	5.37	27.75	2.54	1.48	2.68	16.31	...	63.29
XXI.—Other miscellaneous machinery	...	...	...	...	...	...	...	...	...
XXII.—Pipes connecting to XXI	2.62	.53	...	...	...	.94	8.53	...	12.62
Total without water	139.11	677.39	50.92	359.75	278.97	358.48	33.43	...	1,898.05
XVIII.—Water	2.15	56.13	.29	42.97	28.49	42.59	.64	...	173.26
Total with water	141.26	733.52	51.21	402.72	307.46	401.07	34.07	...	2,071.31
XIXa.—Government stores	...	...	...	...	...	...	...	40	40.
Total	...	...	...	...	...	...	...	...	2,111.

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designs and plans from the same plans and specifications of the Department; where, therefore, there are deviations from these plans and differences between their own designs, these changes and differences may fairly be assumed to be the result of definite and well-considered intention.

The general characteristics of the machinery selected for comparison are as follows: In every case twin screw. Where power is given it is the combined power of both main engines. Where weights are given they are in every case the weights of all objects of that particular kind.

We will take the following vessels:

Four monitors of about 2,000 I.H.P., 250 pounds boiler pressure, water-tube boilers. *Arkansas*, *Nevada*, *Florida*, *Wyoming*.

Four cruisers of about 4,000 I.H.P., 250 pounds boiler pressure, water-tube boilers. *Denver*, *Des Moines*, *Tacoma*, *Cleveland*.

Three battleships of about 10,000 I.H.P., 180 pounds boiler pressure, Scotch boilers. *Illinois*, *Alabama*, *Wisconsin*.

Three battleships of about 16,000 I.H.P., 250 pounds boiler pressure, water-tube boilers. *Maine*, *Missouri*, *Ohio*.

Two armored cruisers of about 23,000 I.H.P., 250 pounds boiler pressure, water-tube boilers. *Pennsylvania*, *Maryland*.

Three destroyers of about 8,000 I.H.P., 250 pounds boiler pressure, water-tube boilers. *Chauncey*, *Paul Jones*, *Worden*.

Before proceeding to a detailed consideration of the machinery weights by groups, let us compare the weights of the main engines alone. Under the term main engines, are included—

Group I.—Main engine cylinders entire group.

Group II.—Shafting; crank shaft.

Group III.—Framing and bearings; bedplates, columns, tie rods and crosshead guides.

Group IV.—Reciprocating parts; entire group.

Group V.—Valve gear; entire group.

The actual weights of these parts are as follows:

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n Engineering, for the be found interesting.

INERY. (ALL WEIGHT

		DESTROYERS.	
	Engines, 3-cylind	Water-tube boilers.	
Scotch. — *Illinois.*	Scotch. ft. — *Alabamay.*	Thornycroft. — *Paul Jones.*	Thornycroft. — *Worden.*
91.03	87.16	20.452	19.967
59.99	59.27	15.547	18.953
73.22	73.85	12.56	13.945
26.58	24.5	4.107	5.614
24.52	24.57	3.568	5.536
23.69	22.96	7.761	7.658
15.95	16.18	2.827	2.306
23.64	19.5	1.498	2.078
331.76	335.94	58.887	62.165
60.86	60.57	17.338	12.777
69.49	60.57	5.267	3.452
40.06	30.827	9.493	10.665
32.57	31.63	9.305	6.842
20.87	28.18	3.272	2.647
35.32	34.62	3.977	3.628
35.39	31.94	8	6.627
27.42	27.953	6.043	3.745
29.97	38 94	.862	1.027
42.92	37.637	2.157	2.934
16.84	10.393	.419	.357
1,082.09	1,057.18	193.34	192.923
196.58	156.26	12.395	14.55
1,278.67	1,213.44	205.735	207.473

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Table II.—ACTUAL WEIGHTS OF MAIN ENGINES ALONE, WITH CRANK SHAFTS AND BEDPLATES.

	I. Cylinders.	II. Crank shafts.	III. Bedplates, framing and guides.	IV. Reciprocating parts.	V. Valve gear.	Total pounds.	Total tons.	Pounds per actual I.P.H.	Total machinery weight.	Size of engines.
	Pounds.	*Pounds.*	*Pounds.*	*Pounds.*	*Pounds.*				*Per ct. of*	
Arkansas	37,099	4,994	19,260	7,874	8,350	77,571	34.66	45.18	13.77	17—26½—40×24
Nevada	38,111	4,955	19,115	8,354	9,548	80,083	35.75	41.23	12.93	do.
Florida	33,349	4,934	18,456	7,814	8,933	73,486	32.81	31.71	14.77	do.
Wyoming	37,447	4,929	19,344	8,591	11,299	82,110	36.66	35.31	13.82	do.
Denver	62,780	10,717	41,662	17,033	16,775	148,967	66.50	26.79	14.95	18—29—35½—35½×30
Des Moines	59,946	10,322	41,739	16,329	15,756	144,142	64.47	28.49	14.26	do.
Tacoma	68,601	10,926	44,105	17,463	16,848	157,943	70.51	30.18	15.95	do.
Cleveland	62,772	10,766	40,698	16,586	16,261	147,083	65.66	32.07	14.35	do.
Illinois	203,914	41,756	132,795	59,500	54,955	492,920	220.05	38.97	17.21	33½—51—78×48
Alabama	195,160	40,860	133,733	54,961	55,055	479,769	214.18	43.32	17.65	do.
Wisconsin	217,839	41,442	147,709	60,743	62,092	529,825	236.53	42.99	18.51	do.
Maine	241,310	45,828	178,289	76,761	55,272	597,460	266	39.27	16.59	38½—59—92×42
Missouri	251,528	57,736	188,216	77,898	55,767	631,145	281.76	40.35	21.40	34½—53 —63—63×48
Ohio	302,011	54,748	224,529	75,198	78,136	734,622	327.96	46.05	23.93	35½—53 —63—63×48
Pennsylvania	364,367	89,178	305,633	106,712	85,761	951,651	424.84	33.98	19.44	38½—63½—74—74×48
Maryland	392,064	76,087	253,203	112,538	85,579	919,471	410.48	33.35	19.82	do.
Chauncey	47,168	7,909	25,873	11,353	10,820	103,123	46.04	12.89*	21.96*	20½—33—38—38×22
Paul Jones	45,816	7,690	22,630	9,203	11,335	96,674	43.16	12.08*	20.98*	do.
Worden	44,727	8,129	23,885	12,575	12,401	101,717	45.4	12.71*	21.88*	23—34—37—37×22

* These figures are based on estimated I.H.P.

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It will be noted by referring to the sixth column in Table II that the weights of engines for vessels of the same class agree only fairly well. In the monitors, the *Florida* is somewhat lighter than the other three. In the cruisers, the *Tacoma* is a little heavy, while in the remaining classes the weights are scattering. With the *Wisconsin* 22 tons heavier than the *Alabama*, and the *Ohio* 61 tons heavier than the *Maine*, while the *Pennsylvania* is 14 tons heavier than the *Maryland*, it would seem that a closer agreement should be reached, considering that these differences in weight are for the *main engines only*.

Taking the total weight of engines as given in column six of Table II, we have:

Class.	Heaviest.	Lightest.	Difference.	Difference in lightest.
	Pounds.	*Pounds.*	*Pounds.*	*Per cent.*
Arkansas	82,110	73,486	8,624	11
Denver	157,943	144,142	13,801	9
Illinois	529,825	479,769	50,056	10
Maine	734,622	597,460	137,162	22
Maryland	951,651	919,471	32,180	3
Destroyer	103,123	96,674	6,449	6

When it is recalled that corresponding vessels of the same class were designed for the same power, there can be no doubt but that 10 per cent. and 11 per cent. is an unnecessarily large variation in weight for similar engines.

Group V.	*Arkansas.*	*Nevada.*	*Florida.*	*Wyoming.*
	Pounds.	*Pounds.*	*Pounds.*	*Pounds*
1. Eccentrics	767.8	1,014	1,002	1,052.75
2. Eccentric straps	1,147.2	1,182	986	1,072.75
3. Eccentric rods	746	702	650	843.5
4. Links and blocks	441	472	424	589
5. H.P. piston valves	156	160	118	249
6. I.P. piston valves	324	300	236	499.5 (4)
8. L.P. valves	1,061 (P)	1,056 (S)	1,078	1,459
9. Valve stems	461	508	466	556
10. Valve stem crossheads	142	174	222	226
12. Suspension links	436.5	612	606	754
13. Reversing gear	2,667.2	3,368	3,145	3,997.25
Total	8,350	9,548	8,933	11,299
Heavier than *Arkansas*	...	1,198	583	2,949

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In the *Arkansas* class the only appreciable differences are in groups I and V. In group I, the *Arkansas*, *Nevada* and *Wyoming* nearly agree, while their average is 37,552 pounds, being 4,203 pounds or nearly 2 tons heavier than in the *Florida*. As to group V, the preceding table will show where the differences lie.

Similarly in the *Illinois* class, though the *Alabama* is light in group IV the discrepancies are primarily due to group V. The weight saved on the *Alabama* in group IV is interesting as showing what saving is made by considering small items. In this case it is mainly in the connecting rods, which weigh:

Illinois, ...	26,585 pounds,
Alabama, ...	22,235 pounds,
Wisconsin, ...	27,183 pounds,

a difference of 4,350 pounds between *Illinois* and *Alabama*. This amounts to 725 pounds for each rod. The rods for all three ships are of the same length and diameter, but in the *Alabama* all unnecessary metal has been cut away and the brasses cored out and lightened. Thus almost two tons of useless metal were saved and this weight could be held in reserve to balance that amount of overweight found necessary, or desirable, elsewhere.

Turning to group V, where there is a difference of 7,140 pounds between the heaviest and lightest, the details of this group follow:

GROUP V, *ILLINOIS* CLASS.

—	*Illinois.*	*Alabama.*	*Wisconsin.*
1. Eccentrics	7,479	8,528	7,983
2. Eccentric straps	9,826	8,672	10,104
3. Eccentric rods	5,524	5,247	4,494
4. Links and blocks	2,526	2,881	3,802
5. High-pressure valves	1,199	1,132	2,174
6. Intermediate-pressure valves	2,185	2,434	3,821
8. Low-pressure valves	4,485	4,259	7,314
9. Valve stems	2,133	2,072	2,036
10. Valve crossheads	1,991	1,871	2,111
12. Suspension links	3,141	2,680	3,218
13. Reversing gear	14,467	15,279	15,033
Total	54,956	55,057	62,090

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Here the difference in weight is noticeable in subjects 5, 6 and 8.

It is interesting to note that in group III the weights of *Illinois* and *Alabama* agree very closely, although the former has the Bureau type of open framing, while the latter has cast inverted Y-frame on the back.

The weights of the engines of the *Maine* class present more points of interest than any of the others. Here are three vessels with engines designed by their builders, and engines which on trial developed practically the same power, as follows:

Maine, ...	15,214 I.H.P.
Missouri, ...	15,642 I.H.P.
Ohio, ...	15,951 I.H.P.

The engines of the *Maine* are 3-cylinder triple-expansion, with inverted Y-back frames and with all valves piston valves; those of the *Missouri* are 4-cylinder triple, with the Bureau type of open framing and all valves piston, while the engines of the *Ohio* are 4-cylinder triple, with inverted Y-back frames, but with slide valves on the I.P. and L.P. cylinders.

In the two 4-cylinder engines, the *Missouri* and the *Ohio*, note how closely the weights agree, both where there is little opportunity to vary the design, as in the crank shafts, group II, and where the pistons are practically the same diameter, as in group IV. It would also seem, from a comparison of the *Maine* and *Missouri*, that for the same power the weights for reciprocating parts and valve gear are practically the same, whether the engines have three or four cylinders, while the crank shafts and framing only vary by an amount which seems to reasonably allow for the differences in length due to the four-cylinder arrangement. On the other hand, in group V, note the increase in weight due to the use of slide valves on the *Ohio*, and no doubt some of the excess in group I on this ship should be charged to them as well.

The engines of the *Pennsylvania* and *Maryland* are the largest we have to consider, and although they have the same

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size cylinders and the same stroke, there is a difference in weight of about 14½ tons.

The weights of the engines, in detail, for these two ships are:

Groups.	*Pennsylvania.*	*Maryland.*	Difference.
I.—H.P. cylinder	61,897	65,505	...
I.P. cylinder	90,831	98,320	...
L.P. cylinders	193,168	207,307.8	...
Relief, drain and safety valves	2,464	3,125.6	...
Receiver pipes	13,348	12,560.8	...
Stuffing boxes	2,659	5,244	...
Total	364,367	392,063.7	27,696.7
II.—Crank shafts	89,178	76,087	13,091
III.—Bedplates	176,638	...	...
Columns, tie rods, frames, etc.	128,995	...	...
Total	305,633	253,203	52,430
IV.—H.P. pistons	3,594	3,628	...
I.P. pistons	6,788	5,534.5	...
L.P. pistons	16,728	14,498	...
Piston rods	11,180	12,638	...
Crossheads and slippers	17,292	18,723	...
Connecting rods	51,130	57,517	...
Total	106,712	112,538.5	5,826.5
V.—Valve gear	85,761	85,579	182

Here the engines of the *Pennsylvania* are about 14¼ tons heavier than those of the *Maryland*, and this extra weight is largely due to the fact that, in developing the Department's design, the *Pennsylvania's* crank shafts were altered by making each of them in four lengths, and interchangeable instead of in two lengths. It is noticeable that in groups IV and V, which would naturally be unaffected by the change in crank shaft, the two engines nearly agree, while in group I the *Pennsylvania's* is much lighter. The result of this change in shaft was that each crank shaft on the *Pennsylvania* had five more flange couplings than on the *Maryland*, and to accommodate these additional couplings each shaft had to be made nearly 3 feet longer. But the change was even more far reaching, for it involved lengthening the bedplates about 4 feet 2

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inches, adding two more main bearings for each shaft, besides increasing the framing. Further, the *Pennsylvania's* bedplates were 3 inches wider in the top flange and 1 inch wider in the bottom flange. An inspection of the above table shows that in this way the *Pennsylvania* not only lost the weight she had gained on the cylinders and reciprocating parts, but over 5 tons additional.

In the three destroyers given we have a fairly close agreement, at least between the *Chauncey* and *Worden*, although the *Worden* engines are somewhat different from the other two. The *Paul Jones* is light in group III, also in group IV.

GROUP III.

	Bedplates.	Columns and tie rods.	Crosshead guides.	Total.
Chauncey	13,910	8,234	3.729	25,873
Paul Jones	11,360	8,157	3,113	22,630
Worden	12,359	7.494	4,032	23,885

GROUP IV.

	H.P. piston.	I.P. piston.	L.P. piston.	Piston rods, crossheads and slippers.	Connecting rods.	Total.
Chauncey	603.5	567	1,391.5	2,961	5,925	11,448
Paul Jones	497	496	1,308	2,712	4,190	9,203
Worden	665	688	1,524	3,747	5,951	12,575

The figures in the eighth and ninth columns of Table II are not of much value for comparison, for in column eight the horsepower and in column nine the total weight varies, with no connection whatever with the weights of engines; still they show, in a general way, what may be expected from the different types.

We have now finished with the weights of the main engines proper, and have considered all of groups I, IV and V. There are still parts of groups II and III to be considered, and, deducting the subjects in these groups already included in main engines, we have remaining the following:

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GROUP II.—(NOT INCLUDING CRANK SHAFTS.)

—	Line shaft.	Propeller and stern-tube shafts.	Thrust shafts.	Shaft couplings.	Coupling bolts and nuts.	Total.	Total length shafting.		Weight per running foot.
	Pounds.		*Pounds*	*Pounds.*	*Pounds.*	*Pounds.*	*Ft.*	*In.*	*Pounds.*
Arkansas	11,027		2,570	646	185	14,528	135		107.6
Nevada	9,268		2,650	696	156	12,770	108	10	117.3
Florida	11,240		2,423	624	159	14,455	109	11	131.5
Wyoming	11,212		2,516	1,014	210	14,952	109	11	136
Denver	7,518	22,784	3,557	1,599	664	36,122	181	10	...
Des Moines	7,925	21,283	3,550	1,415	410	34,583	181		191
Tacoma	7,840	22,319	3,578	751	635	35,123	181	2	194
Cleveland	7,524	21,882	3,529	721	702	34,358	181	9	180.9
Illinois	24,350	48,637	12,378	3,332	3,931	92,628	203	2	456
Alabama	24,500	47,570	12,762	4,048	3,033	91,913	202	9	453
Wisconsin	23,192	49,504	12,404	3,278	3,213	91,591	202	9	452
Maine	26,718	81,251	19,049	8,797	4,784	140,599	206	9	680
Missouri	22,055	62,113	16,810	3,457	4,770	109,205	194	9	561
Ohio	26,495	52,575	16,300	4,665	4,214	104,249	191		546
Pennsylvania	23,680	136,907	28,250	18,183	4,388	211,408	250	5	844
Maryland	27,275	128,105	21,202	19,735	2,430	198,747	249	10	796
Chauncey	24,766		1,565	2,566	465	29,362	199	1	147
Paul Jones	23,213		1,688	1,808	432	27,141	208	6	130
Worden	27,184		1,902	3,929	1,311	34,326	...		...

In certain items in the remaining parts of group III there are great variations between the different vessels, as—

Thrustbearings:	Weight.	Difference between heaviest and lightest.
Arkansas, . . .	6,016.4	2,735
Nevada, . . .	3,281	
Florida, . . .	5,370	
Wyoming, . . .	4,911	
Illinois, . . .	20,160	3,590
Alabama, . . .	19,367	
Wisconsin, . . .	22,957	
Pennsylvania, . . .	39,473	4,160
Maryland, . . .	35,313	
Chauncey, . . .	4,201	1,094
Paul Jones, . . .	3,106.5	
Worden, . . .	3,856	

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Turning to group VI, main condensers, we find that here, as elsewhere, the weights agree only fairly well. For the same power, and practically the same cooling surface, a difference of one or two tons seems too much, particularly when we find such a case as is presented by the *Maine* class, where the smallest condenser is the heaviest. The following table gives the particulars in detail:

ACTUAL SIZE AND WEIGHTS OF MAIN CONDENSERS.

	Cooling surface. One.	Inside diameter.	Length between tube sheets.	Number of tubes in one.	Weight of two.	Weight per square foot of cooling surface.	Difference in weight between heaviest and lightest.
		Ft. In.	*Ft. In.*		*Pounds.*	*Pounds.*	*Pounds.*
Arkansas	1,600	...	10- 9½	905	27,136	8.48	3,956
Nevada	1,612	...	10- 9½	913	29,898	9.27	
Florida	1,612	...	10- 9½	913	27,657	8.58	
Wyoming	1,612	...	10- 9½	913	31,095	9.64	
Denver	3,005	4- 4½	9- 2	2,004	27,556	4.59	1,155
Des Moines	2,992	4- 4½	9- 2	1,996	26,897	4.49	
Tacoma	3,005	4- 4½	9- 2	2,004	27,897	4.64	
Cleveland	3,004	4- 4½	9- 2	2,004	26,742	4.45	
Illinois	7,050	5- 8½	11- 6	3,749	53,084	3.76	2,934
Alabama	7,002	5-10½	11- 6	3,722	51,449	3.67	
Wisconsin	7,006	5-10 1/16	11- 6	3,724	54,383	3.88	
Maine	9,729	6- 3	13- 4	4,460	70,472	3.62	8,335
Missouri	9,381	6- 5	12- 3	4,680	67,539	3.6	
Ohio	9,040	6- 2	11-10	4,769	75,874	4.19	
Pennsylvania	14,329	7- 0	15- 6	5,662	102,056	3.56	4,449
Maryland	14,411	7- 5	14	6,292	106,505	3.69	
Chauncey	3,470	51 3/16	9- 3	2,292	17,643	2.54	490
Paul Jones	3,470	53½	9- 3	2,292	17,387	2.5	
Worden	3,496	44	13	1,642	17,153	2.45	

The sixth column in this table may be found useful.

In group VII, main air and circulating pumps, we have two most important auxiliaries. There is not much opportunity for saving weight in connection with the air pumps, except

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possibly in using a double acting pump, as was done on the *Missouri.*

In the case of the circulating pumps and engines, however, there are wide discrepancies. The work for these auxiliaries is almost always definitely stated in the specifications, giving the number of gallons per minute required to be pumped from the bilge, and, usually, the revolutions per minute as well. One would think that with these particulars fixed there would naturally be a very close agreement between the size of the engines and pumps, supplied by different makers, to do the same work. A glance at the following table, however, will show how the size and weight of these auxiliaries vary:

GROUP VII.—MAIN AIR AND CIRCULATING PUMPS.

	Air pumps.		Circulating pumps.		Circulating pump engines.	
	Size.	Weight of two.	Diameter of runner.	Weight of two.	Size.	Weight of two.
Arkansas	16½ × 8½	3,000	30	2,347	5 — 5 × 6	2,369
Nevada	16½ × 8½	3,089	26	2,068	5 — 5 × 6	1,576
Florida	16½ × 8½	2,989	22	1,649	5 — 5 × 5	1,649
Wyoming	16½ × 8½	3,035	27	1,611	4½ — 9 × 6	2,714
Denver	22 × 10	4,948	36	4,702	5 — 10 × 6	2,137
Des Moines	22 × 10	4,736	30	5,210	5 — 10 × 9	3,333
Tacoma	22 × 10	5,213	29	3,128	5 — 10 × 6	2,748
Cleveland	22 × 10	5,044	36	4,754	6 — 6 × 6	2,706
Illinois	2, 12—2, 25 × 18	18,824	42	7,872	11 × 8	9,038
Alabama	2, 12—2, 23 × 20	18,703	54	10,586	14 × 12	6,984
Wisconsin	2, 14—2, 28 × 18	17,728	36	4,862	8½—13½ × 8	6,379
Maine	2, 14—2, 28 × 18	28,121	54	10,619	7 — 14 × 12	7,705
Missouri	2, 12—2, 18 × 22	22,882	48	10,482	10 — 10 × 10	8,836
Ohio	2, 12—2, 25 × 20	19,201	38	6,339	9 — 17 × 10	10,903
Pennsylvania	2, 14—2, 35 × 18	41,661	54	12,090	11 — 19 × 12	18,525
Maryland	2, 14—2, 35 × 18	41,970	45	13,568	11 — 22 × 10	14,554
Chauncey	12 × 6	4,101	20	856	4½ × 5	856
Paul Jones	12 × 6	4,149	22	982	6 × 5	1,204
Worden	12½ × 6	4,033	20	560	3 × 4	573

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Similarly the table of group VIII, propellers, will show how much difference there can be in the weights of objects supposedly very similar and which have been designed to do the same work.

GROUP VIII.

	Diameter.	Disc area.	Developed area.	Weight of two.	Pounds per square foot of developed area.	Weight for one.	
						Hub.	Three blades.
	Ft. In.						
Arkansas	9	63.62	25.6	6,720.4	131.3	970	2,190
Nevada	9-2.7	66.6	24.03	6,392	133	965	2,042
Florida	8-6	56.74	25.95	8,194	157.9	1,187	2,660
Wyoming	8	50.26	25.8	7,508	145.5	1,182	2,224
Denver	11	88.8	50	18,248	182	2,336	6,210
DesMoines	10-6	86.59	34	11,824	173	1,851	3,769
Tacoma	10	78.54	26.75	10,226	191	1,935	2,701
Cleveland	10-6	86.59	34.74	12,234	176	2,024	3,692
Illinois	16-9	220.35	84.2	52,956	314.5	8,189	16,266
Alabama	15	201.06	76	43,677	287.4	6,250	14,522
Wisconsin	15-6	188.7	68	40,921	300.8	7,702	11,024
Main	16	201.06	79	46,877	296.7	5,880	15,584
Missouri	17	216.98	73.7	54,039	366.5	8,096	17,270
Ohio	16-3	207.39	90	48,382	268.9	8,870	13,394
Pennsylvania	18	254.47	92	63,250	343.8	9,062	20,764
Maryland	17-3	233.7	83.7	63,237	377.7	8,770	20,575
Chauncey	8	48.6	24.9	4,900	98.4	...	...
Paul Jones	7-5	43.2	24	3,356	70	...	...
Worden	8	50.26	22.7	4,655	102.5	...	...

We have now finished with those parts most intimately connected with the main engines proper, and, as the groups to be considered next will be found to vary in weight even more, and for quite different reasons, it may be well here to sum up briefly our conclusions regarding the first eight groups.

Speaking generally, the first thing that strikes one is that there must be great differences in design, or in placing of machinery, to cause such differences in weight as in some cases exist. Why this is so it is difficult to see, and it can have

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little bearing on the performance of the machinery when finished, for all of the vessels considered here had very close trial results in their respective classes. We are forced, then, to charge most of these differences in design either to some consideration outside of the efficiency of the machinery or to the individual ideas of those entrusted with the design of the different parts, and, from the standpoint of the machinery efficiency, neither of these considerations is worthy of weight.

The second noticeable feature is the very much closer agreement in weight between the forged and machined parts than between those that are cast. To a certain extent this must necessarily result and cannot be avoided, for castings made from the same patterns, by different foundrymen, would vary in weight due to rapping of patterns, etc. At the same time this variation would not be enough to account for the really great differences we have seen, and here as well it must be due to difference in design, such as larger and thicker ribs, heavier flanges, or a number of causes of this sort.

Proceeding with the group weights, we will take together groups X, boilers; XI, boiler fittings, and, as with water-tube boilers the subject of lagging is included in group X, for the sake of comparison we will take here that part of group XIV that includes the lagging for the Scotch boilers in the *Illinois*, *Alabama* and *Wisconsin*.

In the following table, in most cases, a comparison by subjects is of little value. It is, of course, well known that among water-tube boilers there are great differences in weight between different types. It may not always be realized, however, how great these differences are, and an inspection of the table should certainly make it clear that, among other considerations of boiler efficiency for a particular case, that of weight should not be entirely ignored. The great excess of weight of Scotch boilers over any of the types of water-tube boilers should exert very strong influence towards the use of the latter type. It should be noted that the specifications for material permit the mills a variation in the weight of boiler tubes of from 5 per cent. above to 3 per cent. below the calculated weight based

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BOILERS.—GROUPS IX, X AND XIV.

Name.	Kind.	Group IX, boilers.	Group X, fittings.	Group XIV, Lagging.	Total dry pounds.	Total dry tons.	Group XVIII, water.	Total wet pounds.	Total wet tons.		Total H. S.	Total G. S.	Weight per sq. ft. of H.S wet.
Arkansas	Thornycroft.	133,609	36,528	...	170,137	75.95	18,800	188,937	84.35	4	9,368	198	20.16
Nevada	Niclausse.	124,054	96,150	...	220,204	98.31	19,760	239,964	107.13	4	8,874	220	27.04
Florida	Mosher.	79,501	32,658	...	112,159	50.07	20,880	133,039	59.39	4	9,600	240	13.85
Wyoming	B. & W.	124,313	61,513	...	185,826	82.96	27,716	213,542	95.33	4	8,800	200	21.11
Denver	B. & W.	214,495	86,180	...	300,675	134.23	46,416	347,091	154.95	6	13,200	300	26.29
Des Moines	B. & W.	215,826	85,533	...	301,459	134.54	46,398	347,757	155.25	6	13,200	300	26.34
Tacoma	B. & W.	215,494	83,982	...	299,476	133.69	46,416	345,892	154.42	6	13,200	300	26.1
Cleveland	B. & W.	215,418	81,516	...	296,934	132.56	46,398	343,332	153.27	6	13,200	300	26.01
Illinois	Scotch.	743,162	136,191	27,674	907,027	404.92	391,200	1,298,227	579.56	8	21,649	684.8	57.96
Alabama	Scotch.	752,376	135,708	38,579	926,663	413.69	313,093	1,239,756	553.46	8	21,691.5	697.5	57.33
Wisconsin	Scotch.	732,456	123,966	26,443	882,885	385.22	331,734	1,214,599	542.23	8	21,205	640	57.27
Maine	Niclausse.	1,153,074	166,796	...	1,219,870	544.58	200,920	1,420,790	634.28	24	53,345	1,353.6	26.63
Missouri	Thornycroft.	594,421	206,793	...	801,213	357.68	97,380	898,593	401.16	12	51,372	972	17.49
Ohio	Thornycroft.	582,346	227,595	...	809,941	361.58	78,624	888,565	396.68	12	53,256	924	16.68
Pennsylvania	Niclausse.	1,572,067	211,555	...	783,622	796.26	234,992	2,018,614	901.17	32	68,528	1,600	29.45
Maryland	B. & W.	1,051,900	384,874	...	1,436,684	641.38	252,000	1,688,684	753.88	16	70,944	1,600	23.8
Chauncey	Thornycroft.	127,678	31,123	...	158,801	70.89	19,166	177,967	79.45	4	17,768	315	10.16
Paul Jones	Thornycroft.	134,143	36,607	...	170,749	76.23	14,260	185,009	82.59	4	17,783	315	10.4
Worden	Thornycroft.	139,250	28,620	...	167,870	74.94	20,301	188,171	84.01	4	17,768	315	10.59

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on the designed dimensions, and as a matter of fact the tendency is nearly always for such tubes to be somewhat heavier rather than lighter. Similarly in boiler and other plates there is a slight variation in weight allowed, and here also the tendency is for the plates to become heavier as the rolls wear.

In connection with the boilers it may be well to call attention to the fact that contractors frequently increase the grate and heating surface of boilers without apparently considering the question of the greater weight which such increases will invariably cause. The question in such cases should be well considered by them, for the responsibility rests entirely with themselves, and, should they deem it absolutely necessary to the fulfillment of the contract, they should take into consideration the increased weight in making their estimates.

In group XI, smoke pipes and uptakes, we have reached one of the most fruitful groups for discrepancies between the weights of sister ships.

There are often great differences between the weights of

GROUP XI.—SMOKE PIPES AND UPTAKES.

—	Uptakes.	Smoke pipes.	Total.
Arkansas	7,583	23,500	31,083
Nevada	20,726	24,800	45,526
Florida	12,534	25,551	38,085
Wyoming	5,144	20,444	25,088
Denver	19,294	40,268	59,562
Des Moines	23,152	37,064	60,216
Tacoma	21,865	38,639	60,504
Cleveland	23,486	45,614	69,100
Illinois	...	...	155,667
Alabama	...	...	135,699
Wisconsin	...	...	132,932
Maine	...	...	331,818
Missouri	...	...	267,589
Ohio	...	...	270,876
Pennsylvania	...	...	367,327
Maryland	...	...	364,699
Chauncey	...	...	15,465
Paul Jones	...	...	11,802
Worden	...	...	7,732

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uptakes and also of smoke pipes, and it is usually due to the difficulty in deciding exactly where to draw the line between where uptake ends and smoke pipe begins; this, however, is a matter of detail and would in no way affect the total of the group, which is the sum of the two subjects.

Though the specifications are usually very definite on these points, shipbuilders vary greatly in their design of uptakes and smoke pipes, in respect to the size, kind, location and number of stiffeners used and even in regard to the thickness of plates.

Then, again, it is not unusual to see long and tortuous uptakes, particularly in cases where, as in the *Denver* class, they run for some distance fore-and-aft, and up through the protective deck before joining at the base of the stack. The preceding table gives the particulars of this group.

The type of boiler used will often affect the uptake weights to a considerable extent. When the gases can be taken from the adjacent sides of a pair of boilers a considerable saving in weight is made over cases in which the gases must be taken from an opening in the top, usually the full width of the boiler. An example of the lighter type is presented by the *Wyoming*. As to the smoke pipes, the following table will indicate, in a general way, how slight variations in construction affect the weight.

—	*Arkansas.*	*Nevada.*	*Florida.*	*Wyoming.*
Height	55 ft. 11¼ in.	55 ft.	58 ft.	57 ft. 3 in.
Inside diameter	6 ft. 3 in.	6 ft.	6 ft. 4 in.	6 ft. 2 in.
Air space	3 in.	3½ in.	3 in.	3 in.
Inside plates	No 10 and No. 7.	No. 10 and No. 7.	Half No. 10, half No 7.	No. 10 and No. 7
Outside plates	No. 12; lower course, No 10.	No. 12; lower course, No. 7.	Half No. 12, half No. 10.	No. 12; lower course, No. 10.
Inner stiffeners	4—3×3 T.	4—3×5 T.	4—3×3 L.	4—3×2 L.
Outer stiffeners	4—3×3 T.	4—3×5 T.	4—3×3 L.	4—3×3 L.
Weight	23,500	24,800	25,551	20,444

Groups XII, XIII and XIV, steam and exhaust pipes, suction and discharge pipes, and lagging and clothing, often show marked differences. In these cases certain differences must exist because of different leads of piping, thickness of valves, number of valves, etc. The location of auxiliary machinery will often affect the quantity of piping, &c., to a marked de-

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gree, and where possible, without sacrificing other things, such machinery should be so located as to require the least possible amount of piping.

In naval work the weight of these groups must always be very great, compared with ordinary merchant work, from the necessity of nearly always providing, in the former, cut outs and duplicate connections and attachments to permit of several ways of doing the same thing.

——	Group XII, steam and exhaust pipes.	Group XIII, suction and disch. pipes.	Group XIV, lagging.
Arkansas	28,991	30,898	5,995
Nevada	27,642	32,411	5,346
Florida	29,084	27,808	5,202
Wyoming	21,696	25,665	6,325
Denver	33,552	40,178	9,887
Des Moines	35,813	56,819	13,715
Tacoma	31,129	46,677	10,009
Cleveland	34,807	65,076	11,191
Illinois	89,769	72,987	67,132
Alabama	77,274	73,244	64,421
Wisconsin	89,556	82,740	46,641
Maine	144,438	116,807	30,682
Missouri	126,702	96,326	25,363
Ohio	105,747	100,009	40,095
Pennsylvania	184,589	141,226	18,519
Maryland	166,266	176,113	33,193
Chauncey	20,446	18,285	5,465
Paul Jones	21,247	20,882	7,335
Worden	23,824	15,382	5,810

In group XV, ladders and gratings, there is sometimes an opportunity to save weight; many ships have too much grating, and often the ladders are of a very heavy type of construction. The open-tread form of ladder, which is now permitted by most specifications, when combined with angle-iron sides, permits of a very light and strong ladder. The old cast-iron treads were heavy and cumbersome, and their use should be avoided. Some of the forms of gratings could be lightened, and there is often an opportunity to save weight in floor plates, particularly in the supports. Following are the weights of this group:

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GROUP XV.—LADDERS, GRATINGS, FLOORS, ETC.

	Pounds.		Pounds.
Arkansas	17,507	*Maine*	76,939
Nevada	19,122	*Missouri*	96,415
Florida	14,333	*Ohio*	126,738
Wyoming	28,861		
Denver	32,348	*Pennsylvania*	96,117
DesMoines	39,540	*Maryland*	113,832
Tacoma	39,341		
Cleveland	41,885	*Chauncey*	8,402
Illinois	79,149	*Paul Jones*	8,917
Alabama	79,015	*Worden*	8,126
Wisconsin	81,942		

Group XVI, auxiliaries, will be given in detail. A close inspection of the table will show that the differences in weight are much greater than might be supposed. Often this is due to the fact that one builder may employ a larger auxiliary for certain work than another; there is, however, considerable difference in weight between auxiliary machinery of the same size built by different makers.

In group XVII, fittings and gear, we also find great differences in weight. The following is a partial table of this group.

GROUP XVII.—(PART) FITTINGS AND GEAR.

—	Feed and filter tanks.	Water-service pipes.	Lubricating gear.	Separators.	Feed heaters.
Nevada	4,386	1,479	1,823	2,025	2,594
Florida	4,052	1,279	1,004	2,129	3,536
Wyoming	4,929	1,337	996	1,975	3,990
Denver	12,679	2,317	1,963	1,521	6,825
DesMoines	11,630	2,111	1,415	1,058	3,420
Tacoma	12,757	3,148	2,034	2,629	3,071
Cleveland	12,449	2,598	2,381	1,252	2,508
Illinois	11,080	3,317	2,272	4,819	10,053
Alabama	9,968	3,336	3,519	7,923	11,866
Wisconsin	17,794	6,484	3,415	2,936	13,674
Maine	21,202	3,309	4,100	12,054	9,143
Missouri	19,799	5,286	3,947	6,179	11,748
Ohio	23,024	7,742	2,794	5,808	17,794
Pennsylvania	21,319	7,141	4,136	12,297	11,859
Maryland	28,484	6,151	4,007	6,537	12,609
Chauncey	1,478	1,037	741	1,279	4,163
Paul Jones	2,777	1,472	747	1,545	2,434
Worden	1,427	503	658	...	2,567

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GROUP XVI.—AUXILIARIES.

——	Feed pumps.	F. & B. pumps.	W. S. pumps.	Other pumps.	Aux. condensers and pumps.	Shaft pumps.	Turning engine.	Ash hoists.	Blowers.
Arkansas	3,328	1,126	1,129	1,427	5,561	630.3	1,341	1,014	5,357
Nevada	3,186	1,510	1,091	1,412	5,590	...	1,788	1,417	4,206
Florida	3,415	914	1,022	2,429	5,566	...	1,143	979	4,866
Wyoming	4,088	1,380	1,374	1,126	4,935	745.5	2,451	1,461	3,170
Denver	9,376	5,699	2,796	2,835	9,226	373	3,472	5,632	7,411
Des Moines	8,992	4,817	2,417	2,403	9,362	542	4,391	4,604	8,064
Tacoma	9,214	5,481	2,762	2,740	8,940	729	2,838	3,134	5,120
Cleveland	9,335	5,833	2,875	2,916	9,198	724	3,495	2,418	2,557
Illinois	19,396	6,552	6,486	...	18,673	6,216	5,598	5,141	11,218
Alabama	20,260	6,761	6,628	67	14,378	3,155	8,333	4,931	10,206
Wisconsin	18,056	6,001	6,016	...	13,439	3,572	8,818	5,012	16,856
Maine	41,155	7,378	7,147	6,872	15,401	3,773	8,896	8,599	19,157
Missouri	28,856	6,334	6,379	6,018	20,196	6,785	5,441	7,266	17,590
Ohio	21,243	6,644	6,666	9,194	15,401	3,663	7,306	8,184	37,587
Pennsylvania	53,184	6,253	...	8,878	15,017	1,600	11,483	11,200	43,196
Maryland	50,175	5,859	...	8,722	12,835	1,382	7,436	8,730	36,154
Chauncey	6,036	1,991	...	...	...	...	740	1,644	4,129
Paul Jones	7,819	2,731	...	...	165.5	...	422	730	2,802
Worden	6,916	2,380	...	...	...	...	706	806	4,034

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Group XVIII—Water. In this group the principal differences in weight are due to difference in size of feed tanks, different types of boilers and difference in piping.

A partial tabulation of this group is as follows:

GROUP XVIII.—(PART) WATER.

—	Water in condensers.	Water in feed and filter tanks.	Water in boilers.
Arkansas	2,200	4,160	18,800
Nevada	2,984	8,126	19,760
Florida	2,556	5,702	20,880
Wyoming	2,812	5,718	47,716
Denver	5,312	19,880	46,416
Des Moines	6,718	16,204	46,398
Tacoma	6,200	21,680	46,416
Cleveland	6,292	19,374	46,398
Illinois	14,790	20,300	391,200
Alabama	13,722	23,362	313,093
Wisconsin	16,146	17,756	331,735
Maine	18,200	...	200,920
Missouri	19,220	...	97,380
Ohio	17,536	...	78,624
Pennsylvania	24,000	59,272	234,992
Maryland	27,916	76,800	252,000
Chauncey	8,525	2,330	19,166
Paul Jones	7,082	3,532	14,260
Worden	7,546	2,320	20,301

GROUP XX.—MISCELLANEOUS MACHINERY.

This group is composed of the independent plants under the Bureau of Steam Engineering, the distilling, heating, refrigerating and air-compressor plants. Some of them agree fairly, but in others, notably the heating plant, there are great differences.

It is believed that if attention was given to the heating plants, with a view to reducing the quantity of piping used, and if the use of cast-iron radiators was avoided, coils being used, a material saving in weights would result.

In the evaporating and distilling plant effort should be made to use as few evaporators and distillers as possible, the

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gain then being not only in these items but in the piping as well. On sister ships using two evaporators compared with those using three, the total capacity being the same in each case, considerable gain is apparent in favor of the smaller installation.

Group XXII is composed of the piping necessary to operate those auxiliaries not under the Bureau of Steam Engineering. Here there is bound to be more or less difference, owing to the different location of these auxiliaries on different vessels, but by care in making the pipe leads as direct as possible, and, whenever it can be done, by working from a single main, an appreciable gain can be made.

Regarding the distribution of machinery weights in different parts of the vessel, the following table, giving averages of the vessels of each class, may be found of interest:

PER CENT. OF TOTAL MACHINERY WEIGHT IN DIFFERENT COMPARTMENTS.

——	Stern.	Engine rooms.	Firerooms.	Bow.
	Per cent.	*Per cent.*	*Per cent.*	*Per cent.*
Monitor class	5.22	37.14	53.27	4.37
Denver class	7	40.18	49.10	3 72
Illinois class	6.56	33.80	58.04	1.60
Maine class	8.34	37.27	52.86	1.53
Pennsylvania class	6.20	33.67	58.88	1.25
Destroyer class	5.77	41.08	52.76	.39

It will be noted here that the percentages in the different compartments agree very well, considering that there are always some items, such as distilling, dynamo and refrigerating plants, the location of which varies somewhat with every design. Of course, the boiler and engine-room percentages will shift somewhat between themselves, depending upon whether light or heavy boilers are used, as illustrated by the *Illinois* class with Scotch boilers.

It should be emphasized that as in naval work the machinery installation is the only element of a vessel on which a penalty is exacted for excess weight, it would seem important to give great attention to this subject of weight during the period of

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design and construction. Further, from the fact that some contractors save weight in one group and some in another, it is plain that in many cases a substantial saving could be made by not losing sight of this important feature in the regular course of design of details, and that it is in the consideration of individual and small items that the best opportunity for weight saving is offered. Of course, considerations much more important than weight must be the deciding elements in design, still it should be recognized that when the machinery has been designed by the Bureau, the specifications written on that design and the contract based on the two, it is not unreasonable to infer that the machinery described is what the Department desires, so that changes involving increase in weight should be made with caution, and not unless there is some prospect of saving elsewhere an amount equal to the probable excess.

It seems to be considered, in a general way, that if machinery is heavier it is better, more durable, less likely to break down; that excess in weight means increase in strength and in endurance, and is not in itself a disadvantage, and in a sense, and up to a certain point, and considering the machinery installation alone, greater weight undoubtedly does add to the strength and to the endurance of the various parts and does make for a better installation, but it is necessary that this excess weight be placed where most needed. If a contractor wishes to add thrust collars (so increasing the length and weight of bearings), if he wishes to have two sizes of connecting rods (thus doubling the number of spares), he should consider the increased weight. If a large thrust bearing is mounted on a heavy, box-shaped sole plate, instead of on one of the usual flat type, though a great deal of weight is added, there is no gain in strength or endurance or efficiency. If cylinder barrels are carried considerably above the tops of liners, with the cylinder covers extending down inside, thus making a double wall, there is no great advantage over the lighter form, though there is greater weight. It is in such unimportant details as these that lack of consideration of weight is shown.

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In the course of developing designs and of construction there are often reasons, and good reasons, why certain items should be increased in weight, but these reasons may very well have nothing whatever to do with the efficiency of the vessel or of her machinery, and a change in design is not necessarily an advantage to the Government. Could each individual case be decided on its merits there would often be no reason why the additional weight should not be employed; but each case cannot be decided on its merits, for an important point in naval work is the mutual relation of the finished weights. To illustrate: A contractor, in getting out the spring bearings for a new ship, finds that he can, with very little change, use old patterns made for a former vessel having a considerably larger shaft; this is perfectly legitimate and in merchant work would not make the least difference. Suppose, however, that, owing to the larger pattern, 250 pounds was added to the weight of each bearing and that each of the twin shafts had three bearings, this would make a total of 1,500 pounds, nearly three-fourths of a ton, in this one item alone. Such an imaginary case as this is cited merely as an example, but multiply the instances of this sort and we would have a vessel that was forced to carry around as freight, during her entire active career, an utterly useless dead weight which might very well have been utilized elsewhere to advantage. Of course, such cases as this, when taken separately in this way, seem trivial and unimportant, but it is for this very reason that they should be guarded against, as in the aggregate the weight so added amounts to a great deal.

It might be urged that in making heavy machinery a builder has paid for extra material for which he receives no compensation, and if the machinery were made heavier consistently, and always with the idea of strengthening and making more durable certain parts, there might be some foundation for this opinion. The great variation in weight, however, in similar parts, shows that it is rarely material is added because the parts are considered weak, and it is not by any means always added in desirable or necessary places. It may be that old pat-

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terns can be altered and used, or that auxiliary machinery, of possibly an unusually heavy type, can be secured at particularly advantageous terms, or it may be that, due to poor design, no attention has been paid to the question of weight. In each of these cases except the last it is clear that the builder has received what might be considered an indirect compensation for the extra material adding weight to the machinery.

In all machinery it is vital to employ weight enough, but on shipboard there are too many other uses to which weight can be applied to admit of its being spread at random or unnecessarily over various parts. A comparison of the weights of similar vessels built by different contractors will often show that in parts employed under the same conditions there will be great differences in weight.

Considering such cases in which certain vessels have parts of their machinery abnormally heavy, experience has not shown that they last any better, work any better, are less free from accidents or less liable to repair than their sister ships whose weights, in these parts, more nearly agree.

The writer does not wish to be considered as saying that, as far as the *machinery installation itself* is concerned, heavy machinery does not add to the strength and endurance, for it undoubtedly does, *up to a certain point;* but a careful consideration of the question has failed to make it clear that extra weight is always so placed as to be advantageous, and, further, it must always be borne in mind that the machinery installation is only one element of the vessel, which, to be considered a success, must harmonize with the design as a whole.

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TEST OF SHELL BOILERS USING THE RONEY STOKER.

By G. W. Baird, Rear Admiral (and Formerly a Chief Engineer), U. S. Navy.

The Postmaster General, to comply with the "smoke law" in the District of Columbia, determined to substitute pea coal for Cumberland coal.

The Roney Stoker had been in use in the (twelve) boilers of the Post Office Department for about ten years. It was built to burn Cumberland coal, with an inclination too steep for anthracite, but, by providing a dam at the lower edge of the grates, and making a change in the dumping plate, the manufacturers claimed the grates would be efficient with the pea coal. The manufacturers were permitted to make these changes without cost to the Government.

(1.) The object of the experiment was to determine whether or not pea coal consumed on the Roney grates in the boilers would furnish sufficient steam to do the work required.

(2.) To determine the economic efficiency of the pea coal so consumed.

(3.) To determine what quantity of pea coal the Department would need for a year's supply.

From the records of the chief engineer of the building it appeared that during the previous year the actual consumption of Cumberland coal was 4,000 tons.

From the following tests it appears that the Cumberland coal is superior to the pea coal in economic efficiency by about 11.3 per cent.

DIMENSIONS OF THE BOILERS (SHELL BOILERS).

Diameter of the boilers, in feet	5
Length of the boilers, in feet	18
Number of furnaces (three boilers)	3
Width of each furnace, in feet	5
Projected length of the grates and dump, feet and inches	5-8
Angle of inclination of the grates, degrees	45
Total area of the grates in the three boilers, feet	85

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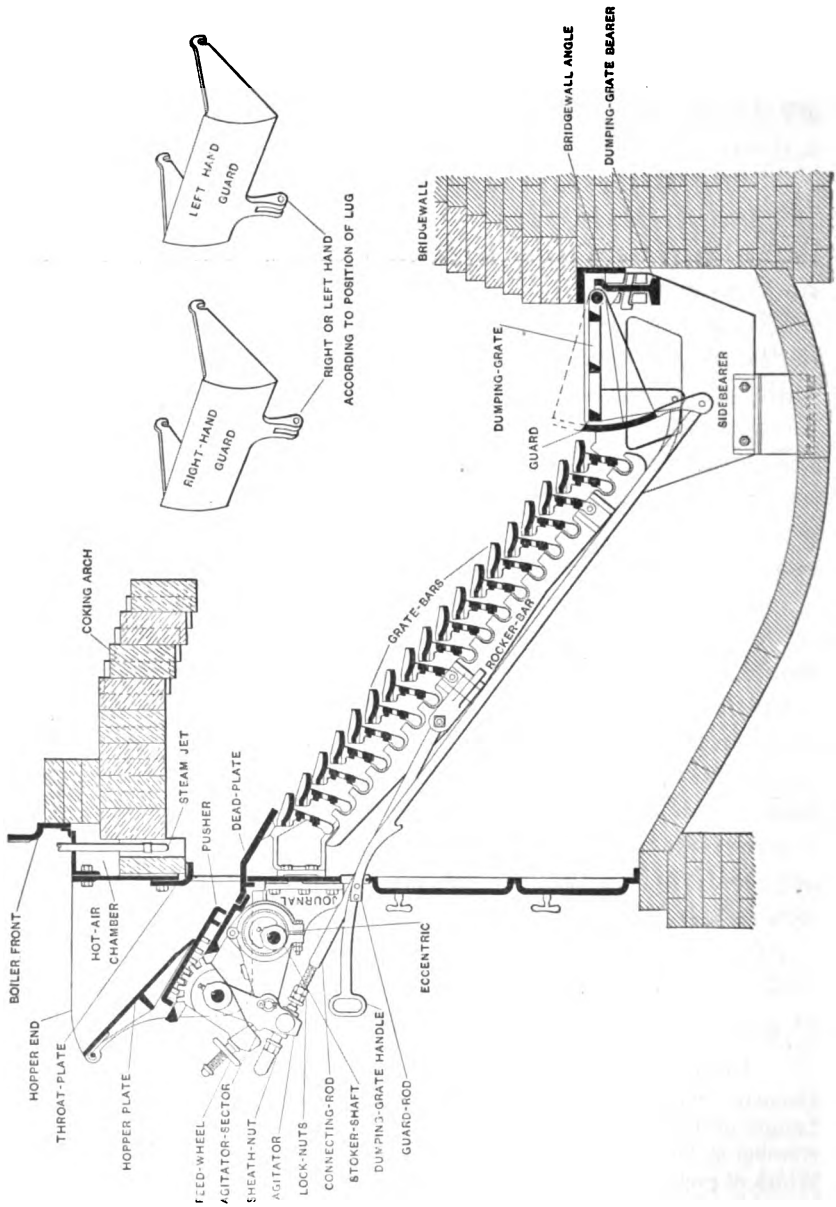

RONEY STOKER.

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External diameter of the tubes, inches	3.5
Length of tubes, feet	18
Aggregate area through tubes for draft, square feet	10.1
water heating surface, square feet	3,373.95
Ratio of heating to grate surface	39.69
grate surface to area through tubes	8.41

The feed pump is a duplex Laidlow pump, having 7.5 inches diameter of steam ends, 4.5 inches water ends and 6 inches stroke of pistons.

The feed-water heaters are of the National make, McKeesport, Pa.

The heaters discharged the feed water into an iron tank which was mounted on a platform scale, and was arranged to hold 1,300 pounds at each charge. This tank delivered into a lower tank from which the feed pump drew its supply. This lower tank was elevated sufficiently to keep a head of water on the pump, as the water was very hot.

There were two experiments, of 12 hours each; one experiment was with the pea coal; the other with Cumberland coal.

The coal and ashes were weighed on a platform scale in the fireroom.

The moisture in the steam was measured by the calorimeter which was invented by Professor C. H. Peabody, of the Massachusetts Institute of Technology, and of the form designed by Mr. G. H. Barrus. After the tests this instrument was "calibrated" for correction.

The temperature of the feed water was taken in the lower tank, close to the feed-pump suction.

The boilers delivered their steam into the main steam pipe, whence it was conveyed to the engines, which continued on their regular duty.

Each experiment was begun at noon and ended at midnight. At 2 P. M. an extra load was thrown on the engines, to ascertain if the boilers would be able to deliver sufficient steam at that load.

	Pea coal.	Cumberland coal.
Duration of experiment, hours	12	12
Total pounds of coal consumed	16,211	16,200
refuse	2,887	1,625
combustible	13,324	14,475

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	Pea coal.	Cumberland coal.
Mean pounds of coal per hour	1,351	1,350
combustible per hour	1,111	1,214
coal per sq. ft. of grate per hour	15.89	17.16
combustible per sq. ft. of grate per hour	13.07	14.28
Total pounds of feed water pumped into the boilers	136,500	167,766
Mean temperature of the feed water	198.6	199.83
steam pressure in boilers, in lbs. per sq. in.	102.416	102.935
lbs. of water vaporized per lb. of coal	8.42	10.35
combustible.	10.23	11.51
Mean pounds of water that would have been vaporized per pound of coal had the feed water been delivered at 212 degrees Fahrenheit	8.87	10.9
Mean pounds of water that would have been vaporized per pound of combustible had the feed water been delivered at 212 degrees Fahrenheit	10.78	12.13
Per centum of moisture in the steam	.18	1.65

The load on the dynamos (and engines) was heavier during the latter experiment, which accounts for the larger consumption of feed water.

It was not found necessary to urge the boilers at any time, though the load was above the average. The tests were made with but 25 per cent. of the boilers in use, and, though there was but little of the (exhaust) steam used in the heating system that day, it is apparent that the boiler plant is sufficient and that the pea coal will give sufficient heat for all purposes.

The purpose of mechanical stokers is not only to save labor, but, by a constant feed and by avoiding the opening of the furnace doors, the inventors and promoters often claim an absolute prevention of smoke. In this respect the Roney seems to be quite as good as any. None of them, however, have diminished the production of smoke sufficiently to satisfy the laws in the District of Columbia; the result is that the Executive Departments are preparing to burn anthracite. The cost of anthracite, compared with that of semi-bituminous coal, is a serious consideration. The pea coal and the buckwheat coal, however, are much cheaper than the furnace size of anthracite, but are less efficient pound for pound. But have the advantage over all bituminous coals of being practically smokeless.

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CONTRACT TRIALS OF THE U. S. GUNBOAT *PADUCAH.*

The *Paducah* is one of the two gunboats provided for in the Act of Congress approved July 1, 1902.

The contract for this vessel was awarded to the Gas Engine and Power Company and Charles L. Seabury & Company, Consolidated, of Morris Heights, New York, on July 6, 1903, the price being $355,000, and time for delivery twenty months. This time was later extended to twenty-four months.

As a detailed description of the *Paducah* and her sister ship, the *Dubuque* appeared in the last issue of the JOURNAL (pp. 589 to 629), the principal dimensions only will be given.

The *Paducah* is a sheathed gunboat of 1,080 tons displacement, with two masts, two funnels, flush main and gun decks and an interrupted berth deck. The principal dimensions and characteristics of the hull are as follows:

PRINCIPAL DIMENSIONS OF HULL.

Length between perpendiculars, feet	174
on L.W.L., feet	174
over all, feet and inches	200- 9
Beam, extreme, feet	35
at L.W.L., feet	35
Ratio of length to beam	4.971 to 1
Depth, molded, main deck, feet and inches	22- 9⅝
Draught, normal, mean, feet and inches	12- 3
mean on trial, feet and inches	12- 3
Displacement, normal draught, tons	1,085.3
trial, tons	1,082.5
per inch at L.W.L., normal draught, tons	10.51
Area midship section to L.W.L., normal draught, square feet	348
L.W.L. plane (draught 12 feet 3 inches), square feet	4,470.7
Wetted surface, square feet	6,900
Center of gravity of L.W.L. plane abaft M.S., inches	3
buoyancy above bottom keel, feet and inches	7- 3
forward of M.S., feet and inches	2-11¼
Transverse metacenter above center of buoyancy, feet and inches	9- ⅞
Longitudinal metacenter above center of buoyancy, feet	194
Coefficient of fineness, block	.509

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Coefficient of fineness of M.S.	.812
L.W.L.	.732
Cylindrical coefficients, longitudinal	.627
vertical	.904
Number of frames	116
Number of compartments	51

ARMAMENT.

Six rapid-fire 4-inch guns; four rapid-fire 6-pounders; two rapid-fire 1-pounders; two Colt automatic.

The ship is propelled by twin outward-turning screw propellers driven by a pair of three-cylinder triple-expansion engines, with cylinder diameters of 9 inches, 15½ inches and 25½ inches in diameter and a common stroke of 21 inches.

Steam is generated by two Babcock & Wilcox marine boilers under a gauge pressure of 265 pounds per square inch, intended to be reduced to 250 pounds (maximum) at the engine.

The boilers have a total of 4,159 square feet of heating surface and 99.78 square feet of grate surface.

The vessel has neither protective deck nor armor of any kind.

The closed-fireroom system of forced draft is used; the air pressure being supplied by two 40-inch fans, each driven by a 3×3-inch twin-cylinder simple engine.

The propellers are of the three-bladed, true-screw, solid-wheel type, having a mean pitch of 6 feet 10 inches; the blades are bent back 8 inches at the tips.

STANDARDIZATION TRIALS.

Runs for the purpose of standardizing the screws were made over the measured-mile course in Narragansett Bay on August 15, 1905; these were commenced at 8·00 A. M. and finished at 10·40 A. M. The weather was thick, rainy and squally with a smooth sea.

The draught taken before and after the runs was as follows:

	Before.	After.
Forward, feet and inches	12– 6	12– 6
Aft, feet and inches	12– 2	12– 1½
Mean, feet and inches	12– 4	12– 3¾
Corresponding displacement, tons	1,093	1,091

The data for these tables will be found on tables I and II.

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Table 1.—U. S. S. *PADUCAH*.

Data from Progressive Runs over Measured-Mile Course off Newport, R. I., August 15, 1905.

Number and direction of run.	Time on course.	Revolutions of main engines. To make 1 knot. Starboard.	Revolutions of main engines. To make 1 knot. Port.	Revolutions of main engines. Per minute. Starboard.	Revolutions of main engines. Per minute. Port.	Revolutions of main engines. Per minute. Mean.	Speed in knots per hour.	Indicated horsepower. Starboard engine. H.P. cylinder.	Indicated horsepower. Starboard engine. I.P. cylinder.	Indicated horsepower. Starboard engine. L.P. cylinder.	Indicated horsepower. Starboard engine. Total.	Indicated horsepower. Port engine. H.P. cylinder.	Indicated horsepower. Port engine. I.P. cylinder.	Indicated horsepower. Port engine. L.P. cylinder.	Indicated horsepower. Port engine. Total.	Indicated horsepower. Total both main engines.	Indicated horsepower. Auxiliaries in operation (estimated).	Indicated horsepower. Total all machinery.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
1-N	7-23.8	874 15	950.45	118.18	128.50	123.34	8.111	19.31	26.49	19.13	64.93	23.98	32.24	35.36	91.58	156 51	25.15	181.66
2-S.	6-39 5	1,017.95	1,017 6	152.88	152.84	152.86	9.011	47.22	55.79	38.77	141.78	28.97	43.33	53.19	125.49	267.27	26 18	293.45
3-N.	6-00 9	982.8	986 75	163.39	164 05	163.72	9.975	63.65	71.2	55.1	189.85	54.05	63.79	77.45	195.29	385 14	25.62	410.76
4-S.	6-51	973.7	979.65	166.44	167.46	166.95	10.256	68	76.51	58.37	202 88	57.47	64.46	65.96	187 89	390.77	27.24	418.01
5-N.	5-58.2	103.8	1,039 6	173.87	174.14	174	10 05	72.74	81.6	60.98	215.32	77 97	99.07	85 97	263.01	478.33	30	508 33
6-S.	5-01	1,002	1,004 6	199.73	200 25	199 99	11.96	118.80	133 9	112.62	365.32	114.53	137.32	126.95	378.8	744.12	31.45	775.57
7-N.	5-12.8	1,099.95	1,100.85	210 99	211.15	211.07	11.509	128.94	158.63	147.99	435.56	131.81	166.24	157.22	455.27	890.83	30.39	921.22
8-S.	4-24.7	1,002.7	1,001.5	227 28	227 02	227.15	13.6	166.02	209.54	206.94	582.5	160.27	207.65	218.04	585 96	1,168.46	34 07	1,202.53
9-N.	4-52.7	1,172.8	1,173 8	240.41	240.62	240.51	12.229	194.47	242.91	235 82	673.2	170.65	246.57	276.53	693.75	1,366.95	37.31	1,404.26

Table 1-A.—STANDARDIZATION TRIALS U. S. S. *PADUCAH*.—HORSEPOWER OF AUXILIARIES.—AUGUST 15, 1905.

No of run.	Circulating pump.	Main feed pump.	Auxiliary feed pump.	Bilge pump.	Flushing pump.	Two blowers.	Dynamos.	Steering engine and hotwell pumps.	Aggregates of power.
1	S. 5.2, P. 4 1	3.6	...	1	0.5	.55	12.15	1	29.1
2	S. 6.3, P. 3.9	3.5	...	1	0.5	.56	12.32	1	28.08
3	S. 4.7, P. 2.2	5.1	...	1	0.5	.7	12.15	1	26.35
4	S. 4.7, P. 2 9	5.8	...	1	0.5	.62	13.02	1	28 54
5	S. 5.9, P. 2.8	6.2	...	1	0.5	2.1	12.84	1	31.34
6	S. 6.5, P. 2.7	7.75	...	1	0.5	1.3	12.91	1	32.66
7	S. 4.7, P. 2.3	8	...	1	0.5	2	12.67	1	31.17
8	S. 4.7, P. 3.5	8.25	...	1	0.5	4.9	12.84	1	35.69
9	S. 4.7, P. 3.8	10.6	...	1	0.5	5.1	12.84	1	39.14
	S. 5.3, P. 3.1	...	...	...	...	...	...	...	...
Averages.....	4.2	6.53	...	1	0.5	2.047	12.637	1	...

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Table No. 2.—U. S. S. *PADUCAH.*

Data from Progressive Runs over Measured-Mile Course off Newport, R. I., August 15, 1905.

Number of run.	Time at beginning of run A. M. (approximate).	Which engine.	Steam pressures. In boilers above atmosphere.	Steam pressures. At engine above atmosphere.	Steam pressures. H.P. steam chest, absolute.	Steam pressures. In first receiver, absolute.	Steam pressures. In second receiver, absolute.	Mean effective pressures. H.P. cylinder.	Mean effective pressures. I.P. cylinder.	Mean effective pressures. L.P. cylinder.	Mean effective pressures. Referred to low pressure.	Vacuum, inches.	Cut-off in cylinders from beginning. H.P., I.P., L.P.	Temperatures. Outside air.	Temperatures. Engine-room working platform.	Temperatures. Fireroom working level.	Temperatures. Injection.	Temperatures. Discharge.	Temperatures. Hotwell.	Temperatures. Feed.	Revolutions or double strokes. Circulating pump.	Revolutions or double strokes. Main-feed pumps.	Revolutions or double strokes. Dynamo engine.	Revolutions or double strokes. Hotwell pump.	Revolutions or double strokes. Sanitary pump.	Revolutions or double strokes. Bilge pump.	Revolutions or double strokes. Auxiliary condenser pump.	Revolutions or double strokes. Fireroom blowers.*	Air pressure, in inches of water.	Pyrometer, base of after stack.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
1	8-05	St'b'd	182	158	63	23	7.5	25	11.7	3	10.18	24.8	Full.	65	90	100	67	76	94	162	415	21	404	48	20	52	...	150	0	300
		Port	185	150	75	25	9	28.55	13.1	5 1	13.2	25					66	80	81	162	380			42				153		
2	8-22	St'b'd	222	150	102.5	25	10.4	47.25	19.05	4.7	17.19	25.5		65	87	100	66	80	97	162	438	20	404	40	24	52	...	151	0	340
		Port	225	215	107	35	12.5	29	14.8	6.45	15.28	25.2					66	84	82	180	376			44				155		
3	8-44	St'b'd	222	213	127	40	12.3	59.5	22 75	6.25	21.54	25.5		68	90	105	66	84	104		400	30	408	28	11	52	...	210	0	375
		Port	228	212	125	40	14	50.4	20.3	8.75	22.06	25.5					66	91	91	162	308			20				215		
4	9-05	St'b'd	260	248	138	43	13.2	62.5	24	6.5	22.59	25.3		62	90	105	68	85	98		400	34	408	48	12	60	...	200	0	400
		Port	265	236	120	40	12.5	52.5	25.3	7.3	22.64	25.8					68	92	90	176	332			19				210		
5	9-22	St'b'd	271	250	138	44	13.5	64	24 5	6.5	22.95	25.3		64	90	109	66	83	116		430	36	408	32	13	60	...	450	0	495
		Port	275	260	165	55	15	68.5	29.7	9 15	27.99	25.5					66	91	90	162	328			15				460		
6	9-42	St'b'd	271	245	197	64	19.5	91	35	10.45	33.93	25.5		63	90	109	68	91	95		441	39	409	20	13	60	...	350	0	410
		Port	274	242	189	68	19	87.5	35.8	11.75	35.06	...					66	100	92	164	326			13				310		
7	10-03	St'b'd	260	248	207	73	22.2	93.5	39.25	13	38.26	25.4		62	91	111	67	96	110		400	40	408	16	12	60	...	450	.15	460
		Port	265	238	190	70	21	95.5	41.1	13.8	39.96	25.5					66	107	96	145	318			18				420		
8	10-24	St'b'd	275	255	247	93	29	111 75	48.13	16.88	47.51	25.5		61	92	111	67	105	118		400	41	408	16	12	64	...	550	.15	510
		Port	277	255	227	95	29	108	47.75	17.8	47.83	25.2					66	104	102	148	360			13				550		
9	10-43	St'b'd	281	258	264	106	34	123.75	52.75	18.18	51.9	24.7		61	93	107	66	114	128		400	58	408	36	12	60	...	970	.75	490
		Port	288	260	257	105	34	108.5	53.5	21.3	53.43	24.6					66	113	111	150	372			30				970		
Averages		St'b'd	248	...	...	...	...	...	...	...	...	...		...	...	...	...	...	...	...	...	...	...	...	...	...	...	387	...	...
		Port	254	...	...	...	...	...	...	...	...	...		...	...	106	..	...	...	...	...	...	...	...	..	...	...	...	...	...

*On all runs forced-draft blowers ran very irregularly.

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THE FOUR-HOURS FULL-SPEED TRIAL.

The contract full-speed trial was commenced at sea, at 12·30 P. M. on August 16, 1905, and finished at 4·30 P. M. The weather was squally and rainy, with a strong breeze from the northwest and a moderate sea.

The working of the machinery, both main and auxiliary, during the trial, was very satisfactory. No water was used on any bearings and all ran cool.

The boilers steamed freely and furnished easily the amount of steam required to exceed the contract speed of 12 knots per hour by nearly one knot.

The draft before and after the trial was as follows:

	Before.	After.
Forward, feet and inches	12– 5¼	12– 5
Aft, feet and inches	12– 1	12– ¼
Mean, feet and inches	12– 3⅝	12– 2⅝
Displacement, tons	1,087	1,078

A summary of the engine-room data taken during the trial follows:

PERFORMANCE ON PRELIMINARY TRIAL, AUGUST 16, 1905.

Trial began	12·30 P. M., August 16, 1905.
Duration of trial, hours	4
Steam pressure at boilers (per gauge), pounds	242.8
H.P. steam chest, S. engine (per gauge), pounds	223.6
P. engine (per gauge), pounds	218.2
1st receiver, S. engine (absolute), pounds	94.1
P. engine (absolute), pounds	103.6
2d receiver, S. engine (absolute), pounds	30.1
P. engine (absolute), pounds	32.9
Vacuum, inches of mercury S. engine	25.66
P. engine	25

MEAN EFFECTIVE PRESSURES IN CYLINDERS IN POUNDS PER SQUARE INCH.

Main engines:

Starboard	H.P., 112.34; I.P., 49.41; L.P., 17.43
Port	H.P., 92.08; I.P., 51.06; L.P., 20.49
Mean pressure in pounds per square inch on L.P. piston, equivalent to aggregate M.E.P. on all pistons, starboard	48.58
Mean pressure in pounds per square inch on L.P. piston, equivalent to aggregate M.E.P. on all pistons, port	49.77

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REVOLUTIONS PER MINUTE.

Main engines.....starboard.....228.58; port..... 230.09
Pumps, air.....starboard, attached; port, attached.
Pumps, circulating.....starboard..... 393.1; port..... 396.4
feed, double strokes per minute 56.1
auxiliary, double strokes per minute, water service.. ..... 14.2
bilge..... 51.6
Hotwell..... .....starboard, 29.8; port 27.1
Blower engines..... 966
Speed of ship, in knots per hour..... 12.823
Slip of propeller, in per cent. of its own speed, based on mean pitch, starboard, 16.76; port, 17.35
Air pressure in firerooms, in inches of water..... .37
Temperature outside air, degrees Fahrenheit..... 59
fireroom, working level, degrees Fahrenheit..... 100

INDICATED HORSEPOWER.

Main engines:
Starboard.....H.P., 167.85; I.P., 216.34; L.P., 214.96; total, 599.15
Port.....H.P., 138.49; I.P., 225.04; L.P., 254.37; total, 617.90
Air pump.....starboard engine, attached; port engine, attached.
Circulating pump.....starboard engine.....4.55; .....port engine, 4.5
Feed pumps, main, one only installed..... 19.6
Bilge pump..... 1
Water-service pumps on sanitary service..... .5
Ventilating engines.....for ship, electric; for engine rooms, electric.
Blower engines for forced draft in firerooms (2), total..... 5.5
Dynamo engines..... 13.
Collective of all main engines..... 1,217.05
main engines, air, circulating hotwell and feed pumps 1,246.7
and auxiliary engines in operation during trial... 1,267.7
all machinery during trial, per square foot of G.S..... 12.705
H.S..... .3048
Main engines, air, circulating, hotwell and feed pumps, per square foot of G.S..... 12.494
Main engines, air, circulating, hotwell and feed pumps, per square foot of H.S..... .2997

COAL.

Kind and quality .....Pocahontas hand-picked, excellent.
Pounds per hour..... 2,698
Pounds per hour, per I.H.P., collective, all machinery in operation 2.129
Pounds per hour, I.HP., collective, main engines, air, circulating, hotwell and feed pumps..... 2.164
Pounds per hour, per square foot of G.S..... 27.07
H.S. ..... .648
Cooling surface, square feet per I.H.P..... 1.071
Heating surface, square feet per I.H.P..... 3.28

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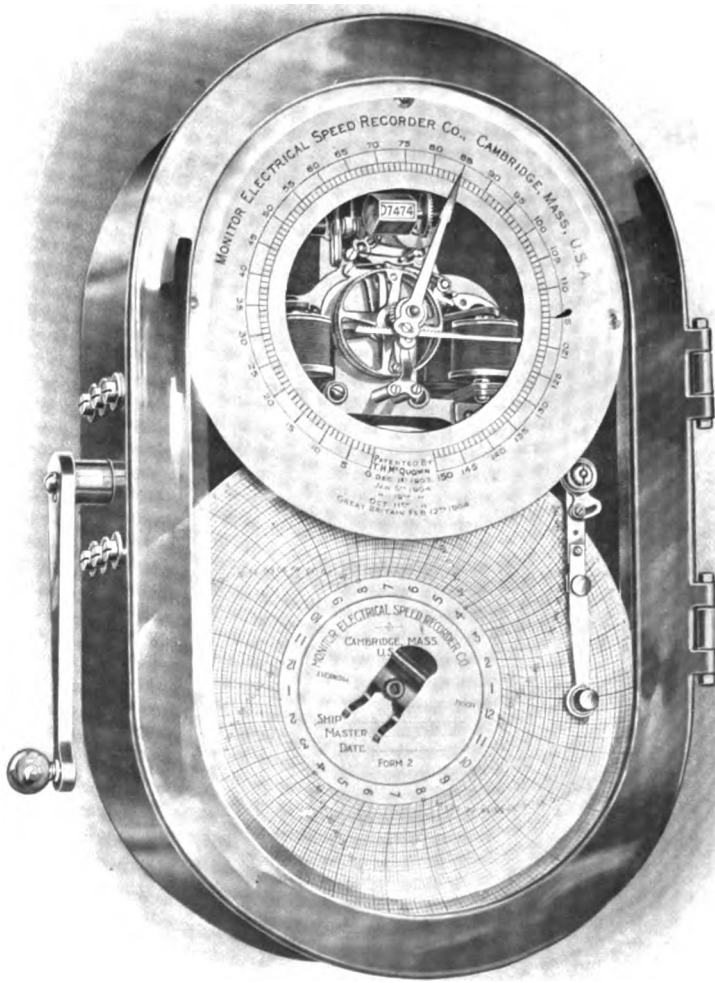

MONITOR ELECTRICAL SPEED RECORDER.

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NEW ELECTRICAL SPEED RECORDER.

By Lieutenant H. C. Dinger, U. S. N., Member.*

The need of an instrument that will indicate accurately at a glance the rate of speed for an engineer located at some distance has long been felt, and numerous attempts at securing an instrument to perform this service have been made.

The Monitor Speed Recorder, a description of which follows, is designed to supply these wants in a simple, practical and feasible manner.

An instrument of this character is of special value to the officer handling a man-of-war in fleet maneuvers. In order to control the speed and movement of vessels engaged in evolutions it is most important to know accurately, at any time, just what speed the engines are making and have just been making, and also to know just when a certain ordered speed is being made.

Besides the speed of the engines, it is important to know the speed the vessel is making at any time without going to the trouble and delay of reading the log between two time intervals. The possession of an instrument that will indicate at any moment the speed of the vessel will be of great help in aiding the gunnery officer to know what orders of lateral deflection of gun's sights to allow when in action or at target practice.

The following is a specification in detail of the working and adjustment of the Monitor Electrical Speed Recorder:

On Figure 1 are shown the principal parts of the Monitor Electrical Speed Recorder.

This instrument is used to show and record the speed of various objects, such as ships by their logs, marine engines, stationary engines, railroad trains, and the velocity of the wind.

* From notes furnished by Monitor Electrical Speed Recorder Company.

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For brevity, we will call the object of which the speed is to be shown the "engine."

The principle of this indicator is this: On the "engine" is an electrical make-and-break that operates a magnet C in the indicator. The frequency of the beats of the armature C1 is governed by the speed of the engine.

The armature C1, carried on a lever C3, by means of a pawl C8, drives a ratchet F3. It is obvious that the rate of speed at which the ratchet F3 is advanced varies with the speed of the "engine."

The "engine" make-and-break works continuously while the engine runs, but the advancing of ratchet F3 is interrupted every thirty seconds by the releasing mechanism which throws the pawl C8 and D7 up off the ratchet F3, which immediately is returned to its starting place by a spring in its hub.

The release is made so that pointer F is left pointing at the speed shown for the last thirty seconds. The release being completed, the advancing of ratchet F3 begins and continues for the next thirty seconds, when another release takes place, and so on.

The hand points at the speed shown for the last full thirty seconds, unless by the speed being increased it is being advanced to a still higher indication.

For ships' logs a one-minute period is used instead of the thirty-seconds period for other purposes.

INSTALLING INDICATORS.

Printed instructions and cuts for installing indicators for various purposes are sent with the indicator, but it should be noted here that it is very important that sufficient electric current be used to be sure of there being no lack of power to drive the magnets. If dry cells, wet cells or storage batteries are used, ascertain the number that will do the work, then double that number in series; this allows for the decrease of strength of the batteries, and then by the diagrams of installation will be seen how the lifetime may be greatly prolonged by wiring the batteries in multiple series. If the current is

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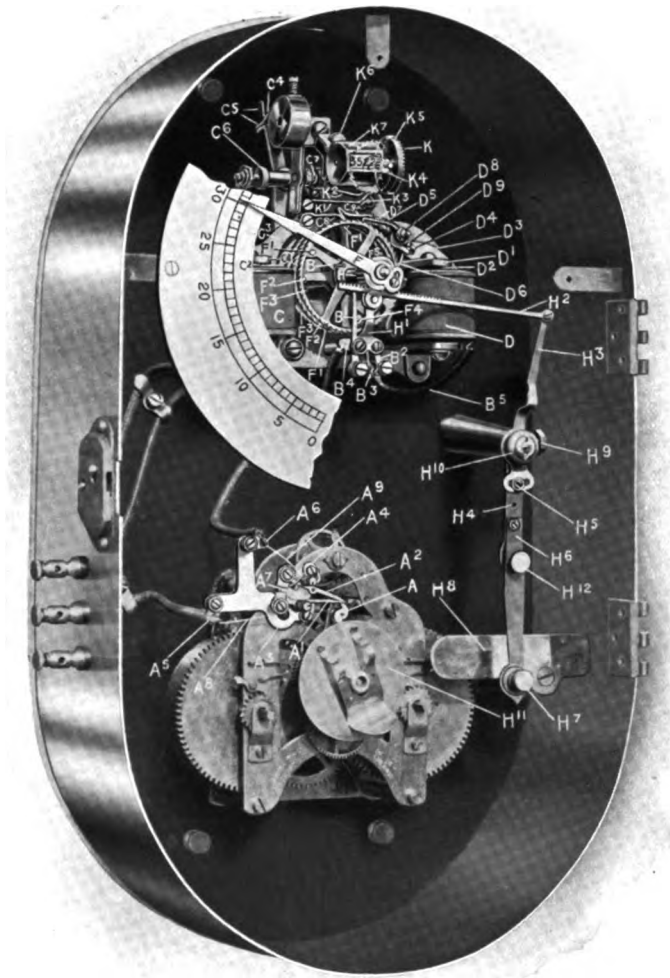

Fig. 1.—Monitor Electrical Speed Recorder.

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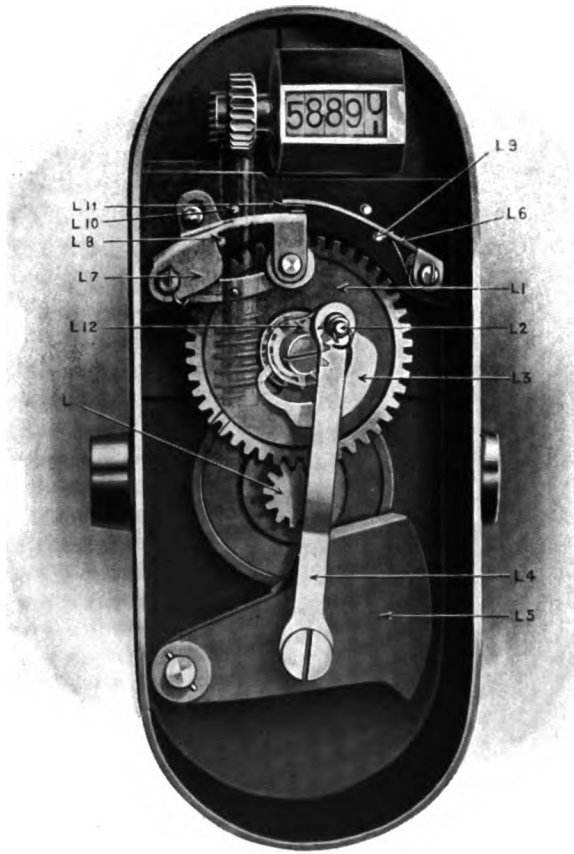

Fig. 2.—TAFFRAIL LOG.

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taken from a dynamo, from 8 to 12 volts for one indicator, and not more than 500 feet of wire No. 18 B. & S. gauge. If a greater length of wire is used a higher voltage may be necessary. For two indicators to one "engine" a current of from 16 to 30 volts is advisable, according to the length of the wire.

The current should be at least 50 per cent. stronger than the amount that will just drive the magnet firmly.

Purpose of Each Part.—The work of each part is as follows: Magnet C, actuated by the "engine," operates lever C3, and by means of a pawl C8 advances ratchet F3 according to the speed of the engine.

Pawl C8 has a ⅂-shaped projection C9, by which it is lifted from the ratchet F3 when D7 is lifted. This projection C9 forms a second face, which engages a tooth of the ratchet ahead of the one engaged by C8; this gives double wearing surface, for F3 is driven by two teeth at a time.

At the top of lever C3 is a weight C4 which retards the quickness of the stroke given to the ratchet F3, and also reduces the noise made. Running directly back from C4 is a link C5 which prevents any severe concussion (such as from gun fire on ship board) from causing weight C4 to throw pawls C8 or K1 off their respective ratchets.

The Releasing Mechanism.—The clock electrical make-and-break, which we call the clock switch, consisting of parts A to A9, operates the magnet D, which causes the release every thirty seconds.

On the seconds arbor of the clock is a two-pointed cam A which revolves with the seconds arbor. The clocks are designed so that the seconds arbor revolves 59 times in an hour, that is, each revolution of the arbor is one minute and $\frac{1}{59}$ part of a minute. The one minute is the time used in advancing ratchet F3 (that is two periods of 30 seconds each) and the $\frac{1}{59}$ part is the time taken to make the release.

Resting on cam A are two levers A1 and A2, so adjusted that A1 falls two ticks of the clock before A2. When A1 falls it closes an electric circuit at A7 which operates magnet D, which causes the release (that is, the lifting of the pawls

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from the rachet). When A2 falls this contact is broken and that release completed.

Armature D1 is given its upward movement by a flat spring D2; this upward movement is stopped by a pin D4 projecting back under the main frame.

When the contact is closed at A7, magnet D draws down armature D1, which is carried on lever D3; this draws down post D9, which is carried on the outer end of lever D3.

This post engages the tails or projections of pawls D5 and D7.

Pointer F and ratchets F1, F2 and F3 are all carried on one center post.

Ratchet F1 is mounted solidly on a sleeve with pointer F and pinion H, so that the three revolve together.

Ratchet F2 is a kind of an idler between the driving ratchet F3 and the pointer ratchet F1.

Ratchets F2 and F3 are turned backward by springs in their hubs, and ratchet F1 is turned backward by a spring in the hub H10 of the recorder arm H3 by means of a rack H2 and pinion H; so the ratchets F1, F2 and F3 will all turn backward when the pawls permit.

When the pawls are in their normal position post D9 holds pawl D6 away from ratchet F2, and pawl D5 is holding ratchet F1 and pointer F at the last indication, and pawl C8 is advancing ratchet F3.

Ratchets F1, F2 and F3 each have engaging pins. Ratchet F3 has a pin that projects through on each side, the inner end engaging screw F4 when F3 is turned back to its starting place.

Ratchet F2 has a pin that projects on each side, the inner end engaging the outer end of the pin in F3 and the outer end engaging the pin in F1. The pin in F1 projects inwardly only.

By this arrangement of the pins F1 is carried ahead of F2, and F2 ahead of driver F3, and by screw F4 they are all prevented from being turned backward beyond their starting place, which is against screw F4.

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The means by which pointer F is left, indicating the speed shown for the last full thirty seconds, while the driving ratchet returns to the starting place, from which it advances for the next thirty seconds, are as follows: When the clock switch closes the contact at A7 magnet D draws down post D9; as D9 starts down, pawl D6 is allowed to engage ratchet F2; this is done in the first half of the downward movement of D9; then in the last half pawls D5 and D7 are thrown up off their ratchets by post D9 engaging their tails, and as the lifting of D7 also lifts C8, ratchet F3 is left free and is at once returned to its starting place against screw F4; also ratchet F1 is held at the new indication by its pin engaging that of F2, which is being held securely by pawl D6.

When the contact at A7 is broken and D9 lifts, it first permits pawls C8, D7 and D5 to engage their ratchets; then in the last half of the upward movement it lifts D6 from F2. F2 at once turns backwards until its pin engages that of F3, then it is carried along ahead of F3 until another release commences.

In the cut the post D9 is shown in the midway position, going either up or down. In this position all the pawls—C8, D5, D6 and D7—are in engagement with the ratchets.

As pawl D5 engages F1 before D6 has released F2 it is obvious that pointer F will be left pointing to the new indication.

The zero switch is a device for cutting the clock circuit when the pointer drops to zero, and so automatically saves the battery. The zero switch is used only on railroads and stationary engines that are stopped frequently; for ships' logs, marine engines and anemometers a hand-operated cut-out beside the indicator is preferred.

On the ratchet F1 is a pin B. When the pointer drops to zero, B engages lever B1, and by a slight movement breaks a contact at B3; this contact is in the circuit from the clock switch to the magnet D.

B4 is an insulated lug at which the heavy wire and light spring wire are connected.

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The cyclometer or revolution counter consists of parts K to K7. On the upper end of the lever C3 is a second pawl K1 which drives a ratchet K. A pawl K2 carried on an adjustable lever acts as a check pawl. Ratchet K is driven forward continuously while the engine runs; it is not interrupted by the "release." To guard against K being stopped in case the clock should stop and so fail to make the release (in which case the pin F3 would make almost a full turn and lodge against the upper side of screw F4 and stop ratchet F3), at F5 two teeth are cut from F3 at a point that will come to pawl C8 just before the pin of F3 lodges against screw F4, so the pawl C8 works freely on this space and keeps the counter K7 running accurately regardless of the rest of the mechanism; this is very important in case of a ship's log when the counter shows the distance traveled.

Ratchet K drives counter K7 by means of a pair of bevel gears K4 and K5. The proportion of these gears varies according to the purpose for which the indicator is intended.

Used as a ship's log, this counter shows nautical miles to one-tenth of a mile, the right-hand figures showing tenths; also on railroad trains and anemometers it shows miles to one-tenth of a mile; but with a marine or stationary engine the total number advanced must be multiplied by ten to get the revolutions made. Applied to side-wheel steamers that never exceed 37 turns a minute, the hand and dial show the turns of the moment to one-fourth of a turn, and the counter shows the revolutions made without multiplying.

The Recorder.—This consists of the record carrier H11 revolved by the clock, and a clip to clamp the records to H11. A table H8 under the pen H7 which is carried by a spring arm H6; H6 is carried rigidly on lever H4 which works rigidly but is adjustable to lever H3.

At the upper end of H3 is pivoted a rack H2 actuated by pinion H. Rack H2 is held to its place with pinion H by a grooved roller carried on an arm H1.

A spring in the hub H10 turns H3 in the direction of the arrow.

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As pointer F advances, pinion H and rack H2 swing lever H3 to the left, and so swing pin H7 to the right on a line of the record corresponding to the figure on the dial indicated by the pointer, and as the clock revolves, the record mark made by the pen shows the rate of speed at any given time.

ADJUSTING THE PARTS.

The Clock Switch.—If a thirty-seconds period is used, cam A will have two points as shown, and, as it is necessary that each period be just thirty seconds, care must be taken not to bend the point of the seconds arbor, for if the two points of A are not true, and one period should be made longer than the next, the result would be that every other indication would be too high and the alternating ones too low.

If a two-point cam is used, lever A2 should fall just two movements of the clock balance (one each way) after arm A1. To make this adjustment accurately, just before A1 drops put your finger lightly on the clock balance, get the right movement so that as you rock it back and forth the escapement will advance with each movement; when A1 drops count the movements until A2 drops; it should be just one movement each way. With a one-point cam A and the one-minute period there should be just two movements each way from the falling of A1 to that of A2.

The adjustment of the lever A3 is mostly perpendicular, and adjusts the distance A1 drops, which should be about half way from the top to the bottom point of A as shown in the cut. If A1 should drop to the bottom of A a good contact at A7 would not be obtained.

The adjustment of the lever A4 is mostly horizontal, and adjusts the length of time between the falling of A1 and that of A2. A8 and A9 are springs that cause the arms to drop, and also serve as conductors of the current of the clock circuit, and so must be insulated from their pivots, and must not touch any part of the clock framework.

Levers A1 and A2 are held to their pivots by split collars.

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These must not be set too closely to the sleeves, for A1 and A2 must be perfectly free and drop quickly.

A5 and A6 are lugs at which springs A8 and A9 are connected to the larger conducting wires.

The contacts at A7 are two pieces of iridio platinum, one of which is cut V-shape. These should come together centrally each way.

Adjusting the Zero Switch.—The zero switch has iridio platinum contacts at B3, the same at the clock switch at A7.

Lever B2 should be set so that when F drops to zero the contacts at B3 will separate about $\frac{1}{32}$ of an inch. The spring on B4 should be just strong enough so that F1 will overcome it when released, say at 2 or 3 on the dial.

Adjusting the Driving Mechanism.—The movement of lever C3 is adjusted by the rubber-tipped bumpers C6 and C7.

First set C6 so that pawl C8 will drop over one tooth and go about one-third way up the next; then set up the check nut *tight.* Then, as the spring C2 advances lever C3 and ratchet F3, bumper C7 should be adjusted so that pawl D7 will fall over one tooth and go about one-third way up the next; then set up this check nut *tight.* Armature C1 should not strike magnet C by a space twice the thickness of spring C2.

Link C5 must work freely, but not permit much side movement of lever C3.

Adjusting the Releasing Mechanism.—Pawl D6 should be formed so that post D9 will hold it about one-sixteenth of an inch from ratchet F2, also so that in the descending it will strike the middle of a tooth as shown in the cut, and not on or just behind the point.

When pawl D5 descends, it, too, should strike in the middle of a tooth, not on or near the point. This on a steady rate of speed permits pointer F to drop back one-half a point as D9 comes down and one-half a point as it goes up at the end of the release. This backward drop of one point is compensated for at the start, where it will be seen by the section of dial shown that there are only four spaces from 0 to 5, but five

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short spaces between each of the other long graduations, so the hand really starts at 1 instead of 0.

The adjustment of the pointer is made by screw F6.

Adjusting Ratchets F1, F2 and F3.—When F3 and F2 are put to place, see that screw F4 is removed and the hub springs set properly to place; their inner ends should hold securely in the keyway of the center post and the outer ends hooked through the slots in their hubs. See that the springs are put in so as to turn the ratchets backward, then turn the ratchets forward until the pins have passed the hole for F4 twice, then put screw F4 firmly to place; let F3 and F2 come back against F4, then put on ratchet F1 with its pin just ahead of that of F2. Put on nut F7, letting the ratchets play just enough to work freely.

The spring used in F3 is similar to a watch main spring 7 wide, 12 strong.

The spring in F2 is much lighter, being .025 ($\frac{1}{40}$) of an inch wide and .005 of an inch thick.

Adjusting the Counter.—The adjustment of the pawl K1 is made with that of C8. Check pawl K2 should be set so that K1 and K2 have equal clearance over their teeth.

Post K3 is to make it impossible for the ratchet K to bounce ahead an extra tooth; to do this K3 must be set so that K1 will lack about one-hundredth of an inch of striking K3. After setting K3 with lever C3 in its forward position as shown, try advancing ratchet K with a gentle pressure direct on K. If K3 permits K1 to raise more than half a tooth then K3 should be set still closer to K1.

By a clamp K6 the counter K7 can be adjusted towards a gear K4 so the gears will engage but will not bind in the least. Never put any oil into the counter.

Adjusting the Recorder.—The grooved roller on H1 should keep rack H2 in easy but sure engagement with pinion H. See that it does not bind at any point of the movement of pointer F.

The spring in the hub H10 is similar to a watch main spring 7 wide, 7 strong. To give this spring its proper tension,

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see that the nut H10 is up tight, and set screw H9 slack, then give nut and post H10 two full turns in the direction of the arrow and hold them there by setting screw H9 up *tight.*

To adjust the pin H7 set down screw H12 until the point of the pen just touches the table H8 all along its course; H8 should be perfectly level; then put a record dial on plate H11 and have the pointer F at some advanced position, say at 30; then by screw at H5 set the pin so it will be on the corresponding line on the record. When the pointer F is at zero rack H2 should not project more than one tooth to the left of H.

If the pen marks too heavily, and it is not due to too much pressure from screw H12, take off the pen cap and insert the end of a small wire or toothpick and gently press the little brass marker out of the pen point a little farther.

Wind the clock once a week; wind both springs. Change the record dials every twenty-four hours. To take up the back lash of the gears the record should be set to the right time by a slight backward movement after the clamp is on. The movement is upward on the pen side.

The pen should be filled once in ten days or two weeks. Do not detach the pen, but simply spring it back an inch or so; remove the cap, fill the pen, using a small dropper, and replace the cap. Use only the ink supplied with this style of pen.

Figure 2 shows the regular taffrail log, make-and-break.

L is the driving gear and L1 the driven. The proportion between these gears varies according to the highest speed at which the engine is to be run, also according to whether the speed is to be indicated once a minute or every thirty seconds. In the cut, L and L1 are equal, and would be used on an engine not exceeding 150 revolutions per minute, the indication being shown once a minute.

For an engine under 150 revolutions per minute, showing the speed every thirty seconds, L would be twice the size of L1.

For an engine under 300 revolutions per minute, showing the speed every thirty seconds, L and L1 would be equal, but

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if the indication was made once a minute L would be half the size of L1. Use 32-pitch gears here entirely.

L1 is a plate, not shown, that engages the inner end of pin L2. L2 projects through a circular slot in cam L3 and draws cam L3 along with it, L3 always following behind L2. L2 is carried on a crank L12, which revolves on the same axis as gear L1 and cam L3. Pin L2 carries a connecting rod L4, which carries a pivoted weight L5. As soon as crank L12 is a little past the upper dead-center, weight L5 throws it to the lower dead center. This throw should begin just before cam L3 engages the roller of L7, and during the throw the cam raises the roller and contact arm L7 and forces the lower iridio platinum contact against the upper one, closing the circuit, also breaking the contact just before the pin L2 comes to the lower dead center. So the contact lasts only for an instant, however slow the engine may be moving. This also makes it impossible for the contacts to be left together when the engine stops.

Adjusting the Engine Switch.—All the parts of this mechanism should work very freely. The weight should begin to fall before cam L3 has commenced to lift the roller, and during the fall the cam should go clear of the roller and break the contact at once.

Contact lever L7 should lift the lever L6 a full one-thirty-second of an inch, and when it breaks the contact they should separate the same distance. Pin L8 is a stop for arm L7, and pin L9 a stop for L6. Pin L10 is to prevent arm L7 from being lifted past L6. L7 should not quite strike pin L10 in its regular lift.

The springs of L7 and L6 should have very light pressure. If the weight L5 does not fall freely enough to make and break the contact instantly, see that the weight and connecting rod and cam L3 and crank L12 are all working perfectly free and that L7 does not strike pin L10; then see that L7 and L6 work so easy that it is impossible to leave the contacts together whenever the shaft of L may be stopped.

The Log Make-and-Break.—This is very similar to that of

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the engine. The counter at the top is run by two pairs of worm gears. All the mechanism above the driving gear is mounted on a plate which is easily removed by taking the binding posts or nuts off the back. To oil the ball bearings remove the box cover, then the worm gears can be given a drop of oil also.

The counter in the log switch shows nautical miles to one-tenth of a mile. To get the log line to run steady a governor of the type supplied should be used. The governors in common use do not obtain steady enough run of the line for successful work with this indicator.

The Anemometer Make-and-Break.—This works on the same principle as the engine switch. Having motion only one way, cam L3 and crank L12 are made in one piece. What has been said about the free making and breaking of the contacts, distance of lift, and clearance in the engine switch holds good in the anemometer.

Packed oil cups are at the top and bottom of the cup shaft. The way to prevent squeaking is to keep it well oiled.

WHAT TO DO IF THE INDICATOR DOESN'T WORK RIGHT.

First. See that the battery is up to the required voltage—at least 8 volts for one indicator, and 16 for two.

Second. See that the wires and connections are free of breaks or short circuits.

Third. See that levers L6 and L7 are making good contact and getting a clear separation.

Fourth. If these are all right, then take off the record, and, without disturbing any adjustments, watch the working of each part of the indicator; and from the instructions already given, the difficulty should be easily located and the proper adjustment made.

Instruments of the above description were installed for test on the U. S. S. *Galveston* and were of great assistance in adjusting the engine speed for the various runs of her official standardization runs at Provincetown.

Sets of these instruments for the engines and also for the

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log were temporarily installed on the U. S. S. *Maryland.* They are now being installed with certain improvements indicated by service on the vessel.

The instruments are found of very material assistance in quickly adjusting the engines to any desired number of revolutions and in indicating variations in the speed due to various causes, such as bad steering, heeling or drop in vacuum. If there is any change the pointer immediately indicates it, much more quickly and accurately than the eye and ear can do so.

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SUGGESTIONS AS TO THE ADAPTABILITY OF A UNIVERSAL STANDARD FLANGE LIST FOR MARINE WORK.

(CONTINUED.)

By Luther D. Lovekin, Associate.

COMPOSITION FLANGES FOR PROPOSED UNIVERSAL STANDARD FLANGE LIST FOR MARINE WORK FOR PRESSURES 0 TO 1000 POUNDS, INCLUSIVE.

(See Drawings Nos. 1, 2, 3.)

To be used on all work where adjoining valves and fittings are of composition or other equally light proportions.

This is suggested so as to avoid the excess weight that would be required if the flanges were made to suit cast-iron valves and fittings. The latter, requiring a much larger diameter of boat circle and consequent outer diameter of flange, will be dealt with under a separate list in this article.

These flanges conform very closely to the ones used by the Bureau of Steam Engineering of the U. S. Navy, for all low-pressure work, and are equally suitable for the Bureau of Construction and Repair of the U. S. Navy, as well as merchant work, under these same conditions.

Owing to the slight variations that exist in the proportions of flanges, builders are required to have on hand various sizes of templets and patterns of flanges, as well as valves and other fittings, all of which involve considerable expense and annoyance to the builders in addition to the various departments in the U. S. Navy.

The object of this article is to bring about a discussion between the various parties interested, so that some step shall be taken to standardize the same.

The question of the strength of these flanges has not been dealt with on account of the very close proximation of this

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suggested list, as previously stated, to the one used for several years by the Bureau of Steam Engineering, U. S. Navy.

This list is one which years of practice has shown to be of ample strength for the work required, while the weight of same is reduced to a minimum, and no further calculations seem necessary. The practical difficulties of attaching flanges to the pipes have to be considered fully when such light proportions are used, and from my experience in this line of work I do not believe we can do better than follow the facts of the case rather than enter into any new theory on these low-pressure flanges.

It is obvious that any malleable material might be used for these flanges, such as cast steel, wrought steel or iron, as well as malleable iron, with perfect satisfaction. When other than brass or bronze composition flanges are used, the brazing of the pipe to the flanges is not essential, provided the pipes are properly expanded into grooved flanges by approved machinery.

I append herewith a list of cast-iron flanges, which are intended for use where cast-iron valves and fittings are used, as previously mentioned.

CAST-IRON FLANGES FOR PROPOSED UNIVERSAL STANDARD FLANGE LIST FOR MARINE WORK FOR PRESSURES 0 TO 100 POUNDS.

(See Drawings Nos. 4, 5, 6, 7, 8, 9.)

This list is particularly well adapted for use where copper pipes are to be expanded into cast-iron flanges, on account of the sectional area of flange not being subjected to the same internal stresses as when wrought-iron or steel pipe is used; it saves weight, and should be encouraged by builders where possible.

For wrought-iron, lap-welded iron or steel pipe, or seamless-drawn steel pipe, it is advisable to combine the thicknesses H & L, and omit bosses on the back of flanges, so as to have more resisting power in the flanges; with sizes above 8 inches diameter, it is necessary, when expanding iron or steel pipe

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into cast-iron flanges, to have the flanges turned on their outer diameter, so as to fit into clamping dies, to prevent flanges cracking when expanding. Hammer tests made from work done in this manner show the flanges to be practically free from any internal stresses; in fact, the metal in the flange seems to become more dense, and improve physically, when under the action of the powerful roller expander necessary to perform this work properly.

Numerous tests made by the author have shown that it is absolutely impossible to expand wrought-iron or steel pipe successfully into cast-iron flanges of the usual proportions without cracking same, unless the flanges are turned on the outside and fitted into dies which have their inner surfaces a counterpart of the outside of the flange; these dies are surrounded by a massive head on the expanding machine, which is greatly in excess of the compressive strength of the material in the pipe, thereby causing the metal in the pipe to flow into any grooves or recesses on the inside of the flange with perfect ease. This gives the flange great holding power, as will be seen from tests made by the U. S. Navy Department, and published in the JOURNAL, A. S. N. E., Vol. XVII, No. 2, 1905, and also "Marine Engineering Journal" of August, 1905.

It will be noted that the cast-iron flanges have been designed for tube dimensions, which vary considerably from pipe dimensions. The tube dimensions retain a constant diameter for the bore of the flange, while the pipe dimensions are determined by the inside diameter of the pipe. This list, however, serves for either tubes or pipes, and the small tables shown herewith give the corresponding pipe measurements that may be used. Thus, if we contemplate using a 12-inch pipe, we use the flange designed for a 12½-inch tube, etc., etc.

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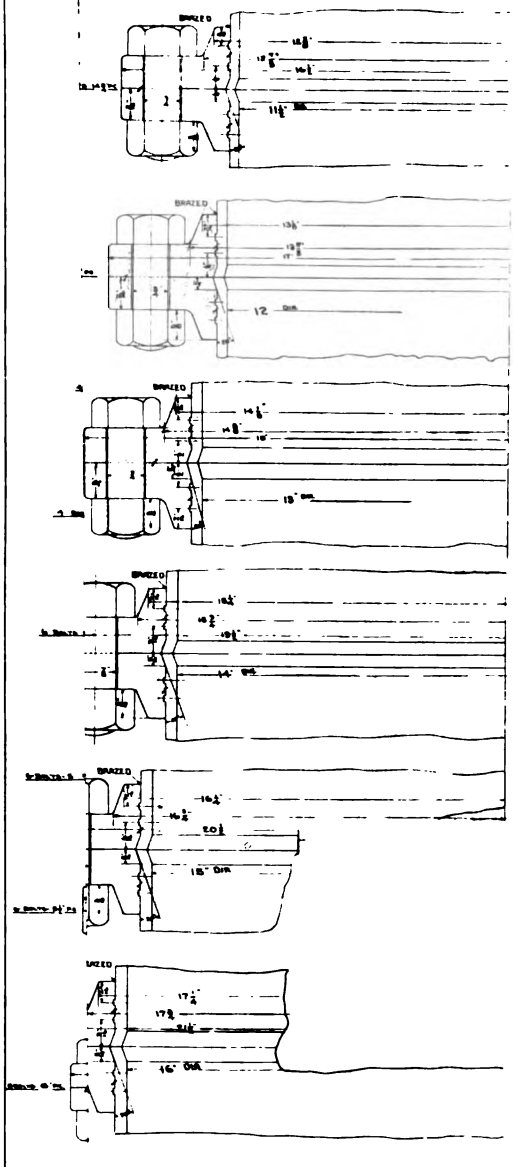

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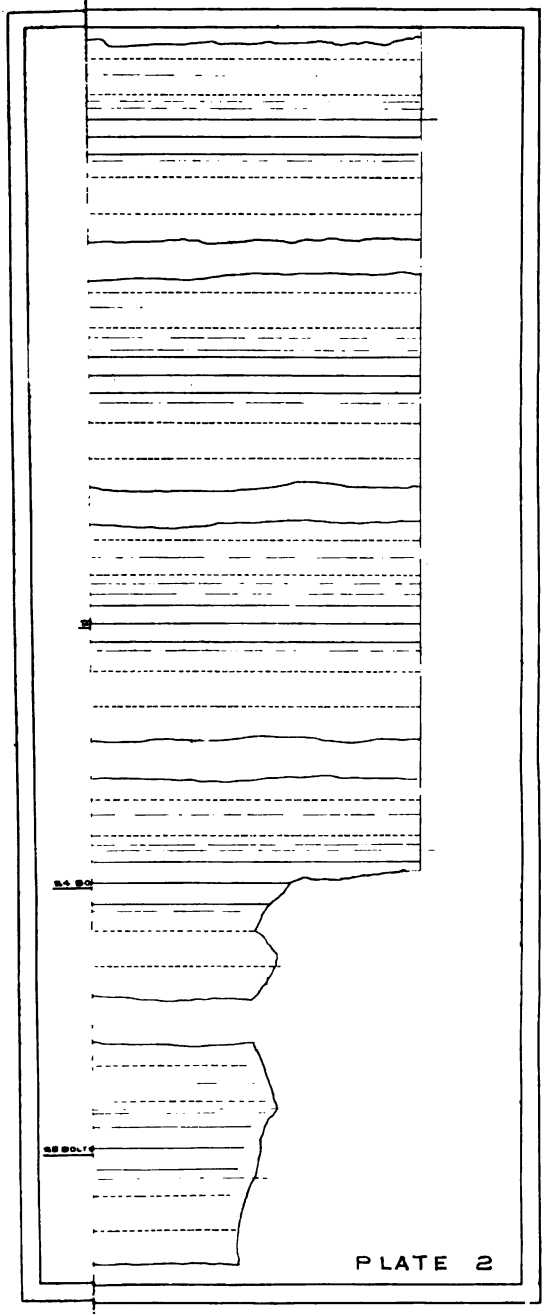

PLATE 2

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PLATE 3.

Wires in Each Frame.

[illegible] PLANKS MAY BE MADE
OF STEEL, CASTINGS, AND THE
FLANGE DEPENDED WITH
REVISED RULES AND MAN-
DET INSPECTION SERVICE.
MADE FOR THE YEAR 1908.

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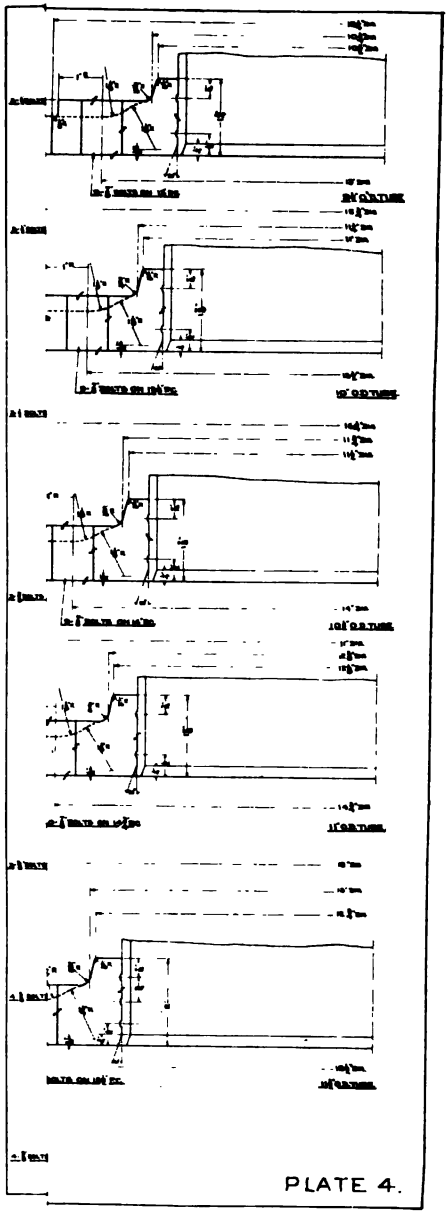

PLATE 4.

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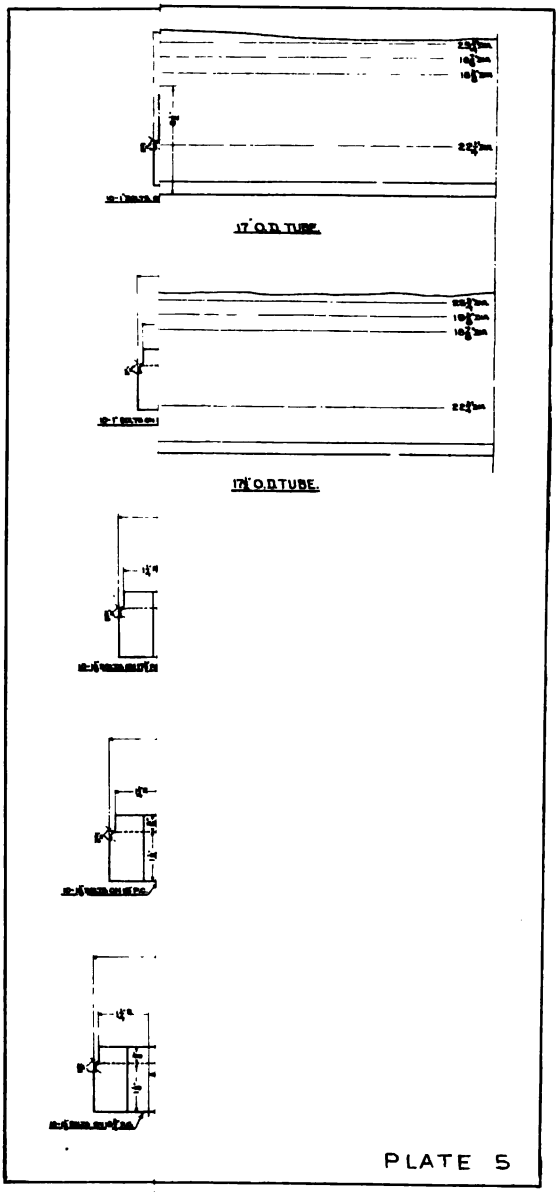

PLATE 5

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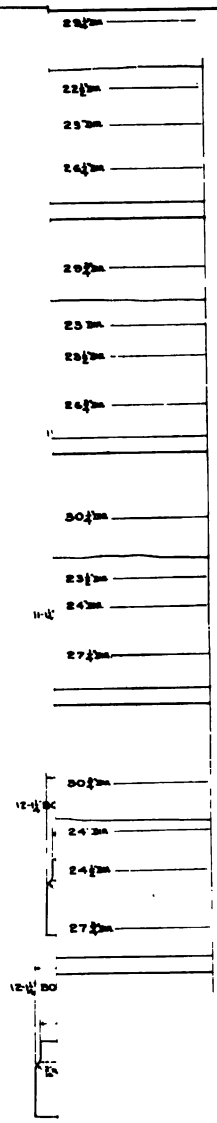

PLATE 6

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PLATE 7.

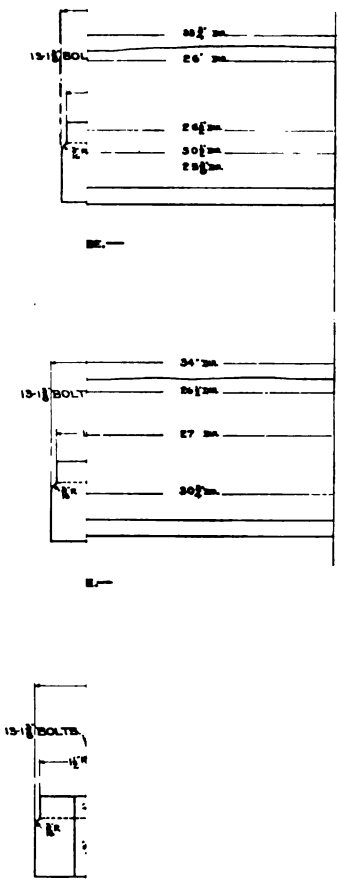

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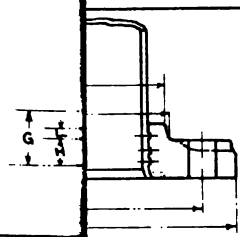
PLATE 9.



E PER SQ. INCH.

INSIDE DIA. OF PIPE	DIA. OF FLANGE B	DIA. OF BOLT CIRCLE C	NUMBER OF BOLTS	M	N	O	P	
1/4	3 1/2	2 1/4	10					
1/2	3 3/4	2 5/8	10					
3/4	4	2 7/8	10					
1	4 7/8	3 1/4	10					
1 1/4	5 1/8	3 1/2	10					
1 1/2	5 1/2	3 7/8	10					
1 3/4	5 3/4	4 1/8	10					
2	6 1/4	4 1/2	10					
2 1/4	6 1/2	4 3/4	10					
2 1/2	7	5 1/4	10					
2 3/4	7 1/4	5 1/2	10					
3	7 1/2	5 3/4	10					
3 1/4	7 3/4	6	10					
3 1/2	8 1/8	6 3/8	10					
3 3/4	8 3/4	6 3/4	11					
4	9	7	11					
4 1/4	9 1/2	7 1/2	11					
4 1/2	9 3/4	7 3/4	11					
4 3/4	10	8	12					
5	10 1/4	8 1/4	12					
5 1/4	10 1/2	8 1/2	12					
5 1/2	10 3/4	8 3/4	12					
5 3/4	11	9	12					
6	11 1/2	9 1/2	13					
6 1/2	12	10	13					
7	12 1/2	10 1/2	13					
7 1/2	13	11	13					
8	13 1/2	11 1/2	13					
8 1/2	14 1/2	12 1/4	13					
9	15 1/4	13						
9 1/2	15 3/4	13 1/2						
10	16 1/4	14						
10 1/2	17	14 3/4						

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SOME STERN FACTS.

By Horace See, Associate.

(Paper read at the Milan meeting of the International Navigation Congress.)

It is remarkable that the natural laws are so often disregarded, particularly in the mechanic arts, and in no branch is it more strikingly illustrated than in the vessel driven by the screw propeller. We might say that here fantastic shapes are the most popular, as shown in the propeller and the stern of a vessel. The propeller, like the Indian's boat, has received a dig from every passer-by, and all sorts of shapes suggested for obtaining the highest efficiency. The expanding pitch with the tip of blade bent aft radially so as to assume a form like a spoon was at one time a prime favorite, but today has given place to the true screw. The latter, after a thorough investigation, was accepted by the writer to be the one, when properly designed, best suited for propelling a vessel, whether the stern was full or lean, as it offered the least resistance in its path whilst obtaining maximum effect. The success with the bronze blade is in the same line, as it is to the thin blade offering less resistance than the thicker iron one, and not to the kind of metal, that this result is due. The frequency, therefore, with which this partiality for the spoon-shaped blade was expressed led the writer, when a contract had been obtained some twenty odd years ago for two vessels similar in every respect, to suggest, in order to solve the problem, that one be fitted with a blade in the form of a true screw and the other with one spoon shaped. These wheels were made solid, the outline as well as thickness of the blades being the same, the only difference being that the one was bent to form a true screw, whilst the other was expanded or twisted to make the spoon shape.

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When the vessels were ready to launch with the wheels in place they were visited by quite a number of persons, from whom the general expression was in favor of the spoon-shaped blade, as it was said by them to take hold of the water better than the true screw. This view was quite true in one way, but not in that desired for obtaining the best results in propelling the vessel, as the one with the true screw was driven at a speed one knot greater than that with the spoon-shaped wheel. This result could not be attributed to the form of the one vessel or the character of its machinery being superior to that of the other, or to the handling of the vessels, but to the wheels themselves, as a new wheel of the true-screw pattern when it replaced the spoon-shaped one brought the speed of this vessel up to that of the other. Again, when the one first fitted with the true screw broke this wheel and had the spoon-shaped one that had been taken off the other vessel placed on it, then her speed also was reduced as in the case of the vessel originally fitted with this wheel.

A tunnel arched longitudinally with its lowest point at bottom of vessel forward of the screw, increasing and attaining its highest point over it and then descending to a point at or below the water line at the stern, is another form that has fascinated many. The United States Government has had a number of vessels of this type built, the contractors for which have lost a considerable sum of money—some having gone into bankruptcy—by failing to secure the contract speed. The original and altered form of stern are shown in Fig. 1. These boats were designed to maintain a speed of 29 knots for two consecutive hours. The *Perry* having attained but 28.2 knots on the progressive speed trials, the contract speed demanded was afterwards reduced to 26 knots, and the duration of run to one hour. The hook at stern was also removed and new propellers placed on two of the vessels. The *Paul Jones* gave the best general results—attaining after the change 28.91 knots on the progressive speed trials and 27.4 on the one-hour's run. The location of propellers and the form of run, still being bad, worked against attaining a higher

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FIG. I.

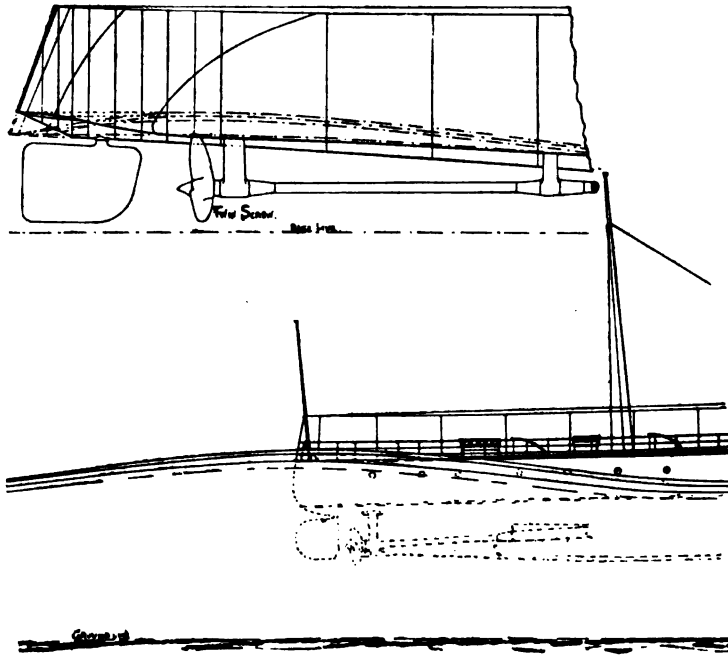


FIG. II.

FIG III

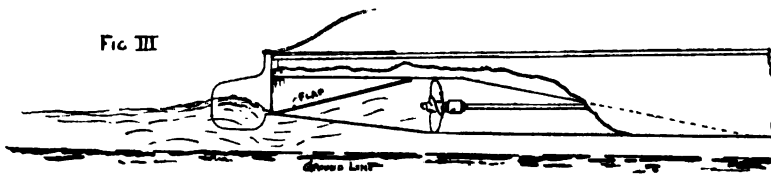
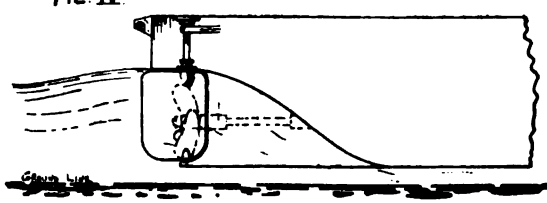


FIG. IV.



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speed. The change of trim in this vessel is not given, but that of the others averaged 5½ feet. The selection of this form of stern is more remarkable in view of the experience in 1889 with the dynamite-gun cruiser *Vesuvius*, contracted to attain a speed of 20 knots per hour, when tried in 4½ to 5 fathoms or comparatively shallow water. The trials, four in number, with this depth of water were unsuccessful. The writer, who was present only on the last one, suggested after that failure a course where the depth of water would not be less than 15 fathoms. This met with favor, and when the next or final trial was made the vessel readily attained a speed of 21.65 knots or 1.65 in excess of the contract requirements. She had extremely fine lines, with a wedge-shaped and not an overhanging stern. On this occasion she stood up without squatting aft and with but very little disturbance in the water, whereas, in the former trials the stern not only was depressed, but a wave was formed with the crest of it so high as to cover the deck at this point, as shown in Fig. II, thereby demonstrating that the water where depth is limited will, when leaving the stern, ascend naturally without mechanical means.

This now brings me to the tunnel stern with an adjustable flap, as shown in Fig. 3. Why a flap at all, is difficult to understand, not only in view of the experience with the above-mentioned torpedo-boat destroyers, but also from the experiments with a shallow-draught launch fitted with a tunnel and hinged flap, of which it is stated in the Transactions of the Institution of Naval Architects, of 1903, page 109: "When the launch is light, drawing 11 inches, with the same power the speed is increased from 9.2 miles an hour with the flap down to 10 miles an hour with the flap up; and when the draught is 28 inches, loaded with 20 tons, the speed is increased from 6.9 miles an hour with the flap down to 8.25 miles an hour with flap up, the power at both speeds being the same. As might naturally be expected, the increase of efficiency due to the lifting of the fan is greater when the boat is loaded, the lower speed in each case being what it would have been if

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there had been no adjustable flap, clearly showing the advantage of the flap." Is not this an admission that the vessel would be better without the flap, meaning the same as the expression that "a dead Indian is the best Indian?" Then why should there be a flap if the boat does better with it raised? Why should it be employed to make the tunnel longer, thereby decreasing the displacement and with it increasing skin resistance? It is also said "that the rush of water being in an inclined direction towards the bottom, rather tends to scour it," but there is a failure to inform us whether the scouring leaves a true or a roughened surface. It is perfectly natural to suppose the latter is the case, and that the flap, not only a disadvantage but also an unnecessary appendage, did project the water downwards to form bridge holes, thereby retarding the vessel in shallow water and demanding greater power to do the work, a result similar to what it would be in the case of a vessel with a roughened bottom, as described in the paper of Capt. Suppan before the Seventh Congress. A screw, however, working in a tunnel with the air from the sides excluded and a free discharge aft seems to be a more desirable formation. It also seems that if the after portion of each side is made to form a rudder, as shown in Fig. IV, an additional advantage is gained in being able to steer a straighter course, as the full surface of the rudders is brought into play, a result not accomplished with the flap, as such a vessel is said to steer badly, the reason, no doubt, being that the water is projected downwards and away from the surface of rudder in proportion to the depression of the flap. With double or twin rudder its position in this arrangement not only affords protection, but also permits the quadrant or tiller being placed below the deck and also in a protected position.

The conclusions drawn by me from the above *in re* the movement of a vessel through a fluid have been as follows:

1. That all vessels create one or more waves.

2. That their character at the stern is partly molded by the shape at this point.

3. That the height of waves above the normal water level

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is governed by the speed of vessel through the water and the latter's depth under the vessel.

4. That the wave at stern in shoal water answers the purpose of a flap, without restricting the flow aft and with least resistance.

5. That the discharge from a screw propeller should not only be free, but without restriction aft of it, no matter what the depth of water or at what speed the vessel moves through it.

6. That discharging the water downward at the stern will scour the bottom and form bridge holes to a greater extent than if discharged directly astern.

7. That the depth of bridge holes will be in proportion to the intensity of the discharge and proximity of the wheel to the bottom.

8. That a tunnel or form of stern directing the water to the screw propeller is valuable when its run limits the amount of skin friction and when it expands aft of the propeller.

9. That the propeller should be placed sufficiently far aft to limit the skin friction from the stream set in motion.

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U. S. BATTLESHIP *VIRGINIA*.

By Lieutenant C. A. Gardiner, U. S. Navy, Member.

The *Virginia* is one of the three battleships authorized by Act of Congress approved March 3, 1899, the others being the *Nebraska*, building at Moran Bros., Seattle, Wash., and the *Georgia*, building at the Bath Iron Works, Bath, Me.

The *Virginia* was built by the Newport News Shipbuilding and Dry Dock Company under contract signed February 15, 1901. This contract provided that the ship should be delivered February 15, 1904, but the time was extended to October 31, 1905.

The total cost of the vessel, excluding armament, certain portions of the armor, etc., is $3,590,000, of which $1,240,000 is for the machinery.

The guaranteed speed is 19 knots per hour, maintained for four consecutive hours, on a mean draught of 23 feet 9 inches, corresponding to a displacement of about 15,000 tons. The usual penalties for failure to attain this speed are attached.

The keel was laid May 21, 1902, and the vessel launched April 5, 1904. The hull is made of basic open-hearth steel, with the frames spaced four feet apart, except under the machinery compartments. The inner bottom extends from frame 13 to frame 95. The double-bottom compartments between frames 60 and 69 are arranged for reserve fresh-water tanks. A coffer dam, 36 inches wide, extends the whole length of the ship on the protective deck, extending 3 feet above the berth deck, packed with corn-pith cellulose forward and abaft the midship armor. All woodwork, except keels, armor backing, etc., is fireproofed. All unexposed decks are covered with linoleum.

The superstructure extends between the two 12-inch turrets, but is not the full width of the main deck, leaving a wide

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gangway on each side. It is recessed on each side just forward of amidships to permit training of the 8-inch turrets, and cut away at the corners to give greater train to the 3-pounder guns on the bridge deck.

BRIDGE DECK.

The bridge deck forms the upper part of the superstructure. Over the forward end is built the forward bridge, on which are the chart house, the emergency cabin, and the entrance to the conning tower. Two 3-pounder, two 30-caliber automatic, and two 30-caliber machine guns are mounted on this bridge. Over this is the flying bridge, with steering and standard compasses and two searchlights. Over the after end of the bridge deck is the after bridge, on which are mounted two 3-pounder and four 30-caliber automatic guns. The following boats are carried on the bridge deck:

Forward on starboard side and on port side, one 36-foot launch, one 30-foot cutter, one 20-foot dinghy, stowed one within the other, and each nest hoisted out by a 5-ton electric crane. Amidships, on starboard side, one 50-foot steam cutter, and aft, on the starboard side, one 16-foot dinghy inside one 30-foot cutter, all three hoisted out by a 15-ton crane. Amidships, on port side, one 36-foot steam cutter, and aft, on port side, one 30-foot cutter, both hoisted out by an 8½-ton crane. Four 3-pounders are mounted forward on this deck, and the same number aft.

MAIN DECK.

This deck extends the full length of the vessel, the superstructure covering it from about frame 32 to frame 78. The two 12-inch turrets with 8-inch superposed turrets, and the two 8-inch amidships turrets, are on this deck, and in the superstructure there are mounted four 3-inch R.F. guns. The hawse-pipes enter this deck through the bow, the anchor chains leading to the windlass house, which projects above this deck. A billboard and an anchor crane for each of the sheet anchors are on this deck. The bower anchors are of the Dunn patent variety, and stow in the hawse-pipes. Two 30-

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foot cutters and two 20-foot dinghies are slung to davits forward, and two 30-foot whaleboats, a 30-foot barge and a 30-foot gig aft. The following are also found on this deck in order from forward: crew's galley, drying room, bakery, officers' galley, paymaster's office, executive officer's office, officers' bath rooms, officers' water closets, wireless telegraph station.

GUN DECK.

On this deck is mounted the 6-inch battery of twelve guns, also four 3-inch R.F. guns forward, and the same number aft. Forward are the crew's shower baths and lavatories, amidships the berthing spaces, and aft the quarters of the Admiral, Captain, Chief of Staff, Executive, Navigator, Chief Engineer, Paymaster, and two wardroom rooms.

BERTH DECK.

On this deck, besides the crew space amidships, there are forward, the sick bay, operating room, isolation ward, dispensary and bath, chief machinists' and chief petty officers' quarters and lavatories, master-at-arms' and sergeant of marines' staterooms, laundry, refrigerating plant and brigs, and aft, the officers' quarters.

PROTECTIVE DECK.

Extends the full length of the ship, and is 3 inches thick on the slope and 1½ inches on the flat. Above it is the cofferdam previously mentioned.

LOWER DECKS.

On the upper platform deck are the magazines, store rooms, sail room, etc., except the 12-inch magazines and shell rooms and forward and after 6-inch shell rooms, which are on the lower platform. In the hold are 6-inch magazines, mine room, etc.

ARMOR.

The side belt is a complete water-line belt 8 feet in width. This belt is of maximum thickness for a distance of 192 feet

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abreast the engines and boilers, 11 inches thick at its upper edge, and maintaining this thickness downward for a distance of 5 feet, from which point it tapers to 8 inches thick at its lower edge. This belt is completed to the bow and stern as follows: for a distance of 60 feet forward and 32 feet abaft the heavy belt above described, the plates are 9 inches in thickness at the upper edge, tapering uniformly to 6 inches in thickness at the lower edge. The next course of plates forward and aft, for a distance of 16½ feet, are 6 inches in thickness at the upper edge, tapering uniformly to 4½ inches at the lower edge. The next course of plates forward and aft, for a distance of 17 feet, are 5 inches at the upper edge, tapering to 4 inches at the lower edge. The belt is completed to the bow and stern with plates of a uniform thickness of 4 inches.

Above the main belt, and covering the central portion of the hull for a distance of 245 feet, the ship's sides are protected by armor of a uniform thickness of 6 inches, extending up to the main deck, this upper and lower-casemate side armor being joined to the barbette of the after 12-inch turret by athwartship armor 6 inches thick, and connected forward by inclined armor 6 inches thick, thus forming a central casemate.

The conning tower is 9 inches in thickness, size, in the clear, 7 by 11 feet, and fitted complete, with space for steering wheel, rudder and shaft indicator, engine-room and other telegraphs, telephones, electric bells, speaking tubes, etc. The signal tower is 5 inches in thickness and 6 feet diameter in the clear.

ARMAMENT.

The 12-inch guns are mounted in pairs in two elliptical, electrically-controlled, balanced turrets, each pair of guns having an arc of fire of 270 degrees. The 8-inch guns are mounted in pairs in four elliptical turrets. Two of these are superposed on the 12-inch turrets, forming part thereof. The other two 8-inch turrets are balanced, electrically-controlled, and placed one on each beam; each pair of guns so mounted having an arc of fire of 180 degrees. The 12-inch turret

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armor is as follows: 12 inches on the slope, 8 inches on the side, 10 inches on the front, and 6 inches on the back. The 8-inch turret armor is 12 inches on the slope, 8 inches on the side, 10 inches on the front, 6 inches on the back, and 2 inches on the top. The 6-inch guns are mounted singly on pedestal mounts, with shields in a central armored casemate on the gun deck, each gun having an arc of fire of 110 degrees. The secondary battery consists of twelve 3-inch R.F. guns, twelve 3-pounders, semi-automatic guns, four 1-pounder heavy automatic guns, four 1-pounder heavy R.F. guns, two 3-inch field pieces, two .30-caliber machine guns, six .30-caliber automatic guns, and four submerged torpedo tubes, two forward and two aft.

Two Barr and Stroud mechanical range-finders are mounted, one in each lower top.

HULL DATA.

Length between perpendiculars, feet	435
on load-water line, feet	435
over all, feet and inches	441-3
Beam, extreme, feet and inches	76-2½
at load-water line, feet and inches	76-2½
molded, feet	76
Ratio of length to beam	5.708
Depth, molded, main-deck side at M.S., feet and inches	41-3⅝
Draught, normal, mean, feet and inches	23-9
Displacement, mean load, tons	14,979
full load, tons	16,094
per inch at L.W.L., tons	61.1
Area of midship section to L.W.L., square feet	1,725
L.W.L. plane, square feet	23,650
Wetted surface, square feet	39,200
Center of gravity of L.W.L. plane aft of M.S., feet and inches	3-3½
above bottom of keel, feet	26.25
buoyancy above bottom of keel, feet	13
forward of midships, feet	105
Transverse metacenter above center of buoyancy, feet	19.18
Longitudinal metacenter above center of buoyancy, feet	519
Number of frames	108
Coefficient of fineness, block	.6696
M.S.	.9618
L.W.L.	.7724

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MAIN ENGINES.

There are two four-cylinder, triple-expansion, inward-turning engines of the vertical, inverted, direct-acting type, in two watertight compartments, separated by a fore-and-aft bulkhead. The order of the cylinders is F.L.P., H.P., I.P., A.L.P. The H.P. and I.P. cranks are at 90 degrees, as are also the two L.P. cranks, and the sequence of cranks is H.P., I.P., F.L.P., A.L.P. The cylinders for each engine are supported by six front and six back columns, flanged and bolted at the bottom to the bed plate, and at the top to the cylinder casings. For each athwartships pair of columns there are two symmetrically arranged diagonal braces, secured at the ends to the columns, and at their centers to each other, and also a horizontal tie rod with sleeve distance piece. The columns are strengthened by horizontal and diagonal fore-and-aft tie rods. All the above of forged steel. The H.P. and I.P. cylinders are so connected as to prevent athwartships motion while permitting fore-and-aft play.

The valves are the single-ported piston type, with cast-iron heads and packing rings and steel followers. The H.P. is an inside valve, the others, outside. The H.P. cylinder has one valve, the others, two each, all arranged to lift out at the top of the valve chest. Each valve has a balance piston working in a cylinder cast with the upper valve-chest cover. Stephenson double-bar link valve gear is fitted, and the I.P. and L.P. valve stems are connected to crossheads.

The floating-lever steam reversing engine with oil-control cylinder is used, and each link is fitted with an independent linking-up gear of the usual pattern, giving a range of cut off of from .48 to .74 the stroke. The valve data is given elsewhere.

Reversing Engine.—Diameter steam cylinder, inches, 16; diameter oil cylinder, inches, 8; stroke, 22 inches.

The cylinders and valve chests are of hard cast iron with cast-iron linings. A jacket space of three quarters of an inch is left between the cylinder casings and their linings. Steam at working pressure enters the H.P. jacket, from which it

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passes through reducing valves to the I.P. and L.P. jackets, each jacket being fitted with a drain pipe leading to an automatic trap.

The H.P. piston is of cast iron, the I.P. and L.P. pistons of cast steel, all having forged-steel followers. The H.P. and I.P. pistons have one cast-iron packing ring, which is kept expanded against the cylinder by springs operating longitudinally on the ends of the ring. The L.P. packing rings are two in number for each piston, and each ring is in eight segments set out by steel springs. The rings break joints and are secured to each other by dowel pins.

The piston rods are of forged steel, hollow, accurately ground and polished, and tapered at each end to fit the piston and the crosshead. The crossheads are of steel, forged with the crosshead pins, and are bolted to the cast-steel slippers. The crosshead guides, both ahead and astern, are of cast steel. The latter are bolted to flanges on the former, and these guides are bolted to brackets on the cylinder casings. The crank-pin brasses are of Navy standard composition, lined with white-metal, the brasses and steel caps secured by through bolts to the connecting-rod ends.

The eccentrics are made in two parts, the larger of cast-iron, the smaller of forged steel. The eccentric straps are of Navy standard composition, lined with white metal, and are secured to the forged-steel eccentric rod by stud bolts.

ENGINE DATA.

	H.P.	I.P.	L.P.
Cylinders, number for each engine	1	1	2
diameter, inches	35	57	66
Stroke	48	48	48
Valves, number for each cylinder	1	2	2
diameter, inches	20	22	25
Valve stems, diameter, inches	2	2	2
Piston rods, diameter, inches	8	8	8
axial hole, inches	4	4	5
length	9 ft. 4½ in.		
Cylinder walls, thickness	2	1⅝	1½
Thickness of liners	1½	1⅝	1¼
Jacket space	¾	¾	¾
Valve-chest liners	2½	2¾	2¾

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Crosshead surface, ahead, square feet	690
backing, square feet	510
Connecting rod, between centers, feet	8
Clearance of all pistons, top, inch	¼
bottom, inch	¾
Crosshead pin, diameter, inches	9¼
length, inches	10½
Crank shaft, number of sections	2
diameter, at eccentric seatings, inches	18½
of axial hole, inches	10¼
coupling disc, feet and inches	2– 8
thickness of coupling disc, inches	3⅝
number of bolts in coupling disc	8
diameter of bolts in coupling disc, inches at center	3½
journals, inches	17½
length of journals, inches	21
forward section, feet and inches	15– 6¼
after section, feet and inches	16–10
pin, length, inches	21½
diameter, inches	18
of axial hole, inches	10½
webs, width, inches	20
thickness, inches	11
Thrust shaft, diameter, inches	16¾
of axial hole, inches	10¼
number of collars on each shaft	12
collars, diameter, inches	27
thickness, inches	2
space between, inches	4
total thrust surface, one engine, square feet	29.335
length, feet and inches	13– 7½
Line shaft, diameter, inches	16¾
axial hole, inches	10¼
length, feet and inches	14–11
after section, feet and inches	14–11½
diameter of coupling disc, feet and inches	2– 8
coupling bolts in each disc	8
diameter, inches, tapering	4 to 3

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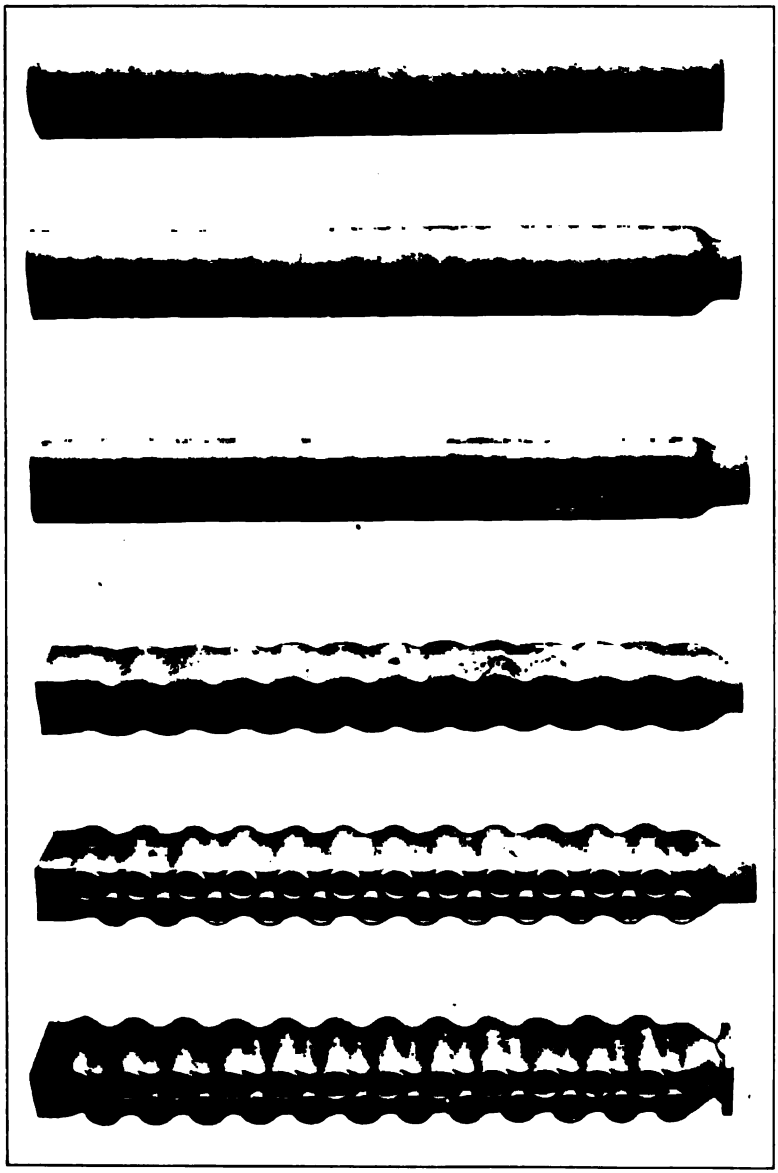

SUCCESSIVE STEPS IN FORMING NICLAUSSE HEADERS.

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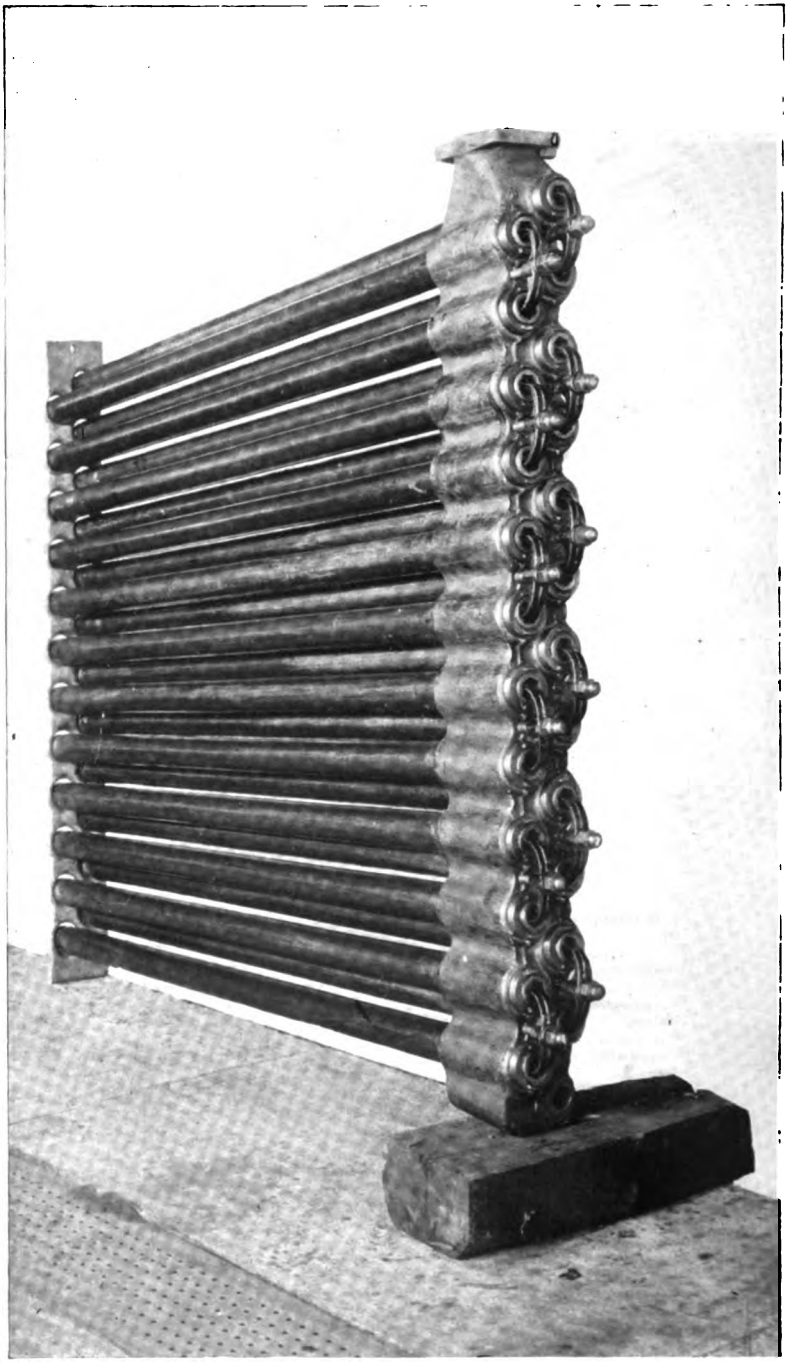

A NICLAUSSE ELEMENT.

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VALVE DATA, U. S. S. *VIRGINIA*.

Engine, inches 35, 57, 66, 66
Valve gear ..... Stephenson link.
Connecting rod, between centers, feet 8

—	H.P.		I.P.		L.P.	
Eccentricity, inches	5		5		5	
Travel of the valve, inches	10		10		10	
Number and size of piston valves, inches	1, 20		2, 22		2, 25	
Side on which steam is taken	Inside.		Outside.		Outside.	
	Top.	Bottom.	Top.	Bottom.	Top.	Bottom.
Width of port, inches	3½	3½	3½	3½	3½	3½
Steam lap, inches	2 5/16	2	2 5/16	2	2 5/16	2
Exhaust lap, inches	—⅛	—¼	—⅛	—¼	3/16	⅛
Angular advance, degrees	36	36	36	36	36	36
Steam lead (linear), inch	⅛	1/16	⅛	1/16	⅛	1/16
(angular), degrees and min.	8, 27	12, 25	8, 27	12, 25	8, 27	12, 25
Cut-off, inches	37 7/16	33 15/16	37 7/16	33 15/16	37 7/16	33 15/16
decimal of stroke	.775	.707	.775	.707	.775	.707
Mean cut-off	.741	.741	.741	.741	.741	.741
Exhaust release, inches	43 5/16	41½	43½	42	44⅞	44¾
decimal of stroke	.903	.865	.907	.875	.935	.925
Compression, inches	4	3	4 7/16	3¼	6¼	5¼
decimal of stroke	.083	.063	.092	.068	.13	.107
Steam opening, inches	2 15/16	3	2 15/16	3	2 15/16	3
Exhaust opening	Full port	Full port	Full port	Full port.	Full port	Full port.
Velocity of steam (piston speed, 960 feet) feet per minute	7,240	6,160	8,660	7,650	10,250	9,050
Velocity of exhaust, valve ports	5,550	5,280	6,690	6,560	7,860	7,770
cylinder ports	5,600	5,300	6,900	6,770	7,725	7,600

Velocity of exhaust to condenser, 24-inch pipe, feet per minute 7,200

BOILERS.

The *Virginia* has twenty-four Niclausse boilers, arranged in groups of four, in six watertight compartments—three on each side of the middle-line bulkhead and inboard of the coal-bunker bulkheads. In each compartment the boilers are arranged in pairs, each pair consisting of one 16-element boiler inboard and one 15-element boiler outboard, the forward pair facing aft and the after pair facing forward, with an athwartships fireroom space between. Passageways run inboard of the boilers throughout the boiler compartments, and communication is established between opposite compartments and between adjacent compartments on the same side by means of watertight doors. The after boiler compartments communicate with the engine room through air locks.

Each boiler element consists of a vertical forged-steel header, with interior diaphragm, and twenty-four inclined generating tubes, fitted with internal circulating tubes, according to the

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Niclausse construction, this arrangement providing 384 generating tubes for each 16-element boiler and 360 for each 15-element boiler, or a total of 8,928. These generating tubes are of steel, seamless drawn, $3\frac{1}{4}$ inches in outside diameter, and No. 6 B.W.G. in thickness, with lanterns integral with the tubes at the header ends, and at the opposite ends, where resting in the rear tube supports, pinched into $2\frac{5}{8}$ inches diameter and capped. The overall length of each tube, inclusive of $9\frac{1}{2}$ inches length of lantern, is 8 feet $3\frac{3}{16}$ inches, and the exposed length 7 feet $5\frac{11}{16}$ inches. The circulating tubes are of No. 21 B.W.G. sheet steel, $1\frac{19}{32}$ inches outside diameter, each formed in two halves, pinched together with folded-over longitudinal joints. The downward inclination to the rear of all tubes is 1 to 10.

An arrangement has been made with the Bureau of Ordnance whereby the compressed air used for the battery is led into the fire rooms and, by means of suitable hose connections, terminating in a pipe with a back-blowing nozzle, may be used to empty the generating tubes of water.

The heating surface of the tubes is 2,291.75 square feet in a 15-element boiler and 2,444.20 square feet in a 16-element boiler. Adding the heating surface of the headers, the total heating surface of a 15-element boiler is 2,320.07 square feet and of a 16-element boiler 2,474.44 square feet. Total heating surface of all boilers 57,534.12 square feet.

The grates in all boilers are 6 feet $8\frac{5}{8}$ inches long, and the width is 8 feet 9 inches for 15-element boilers, and 9 feet $4\frac{1}{4}$ inches for 16-element boilers, making the grate surface for a 15-element boiler 58.78 square feet and for a 16-element boiler 62.86 square feet. The total grate surface is 1,459.68 square feet, and the ratio of heating surface to grate surface 39.41 to 1. The grates are fitted with Navy standard grate bars.

Each boiler has a separate drum, situated directly over the headers and connected to each by means of a double cone nipple, forming Niclausse joints with the bottom of the drum and the top of the header, similar to the joints made by the generating-tube lanterns in the fronts of the headers. These

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nipples are each secured by four steel bolts, connecting the bottom of the drum and the top flange of the header. The drums are 10 feet 1½ inches long in a 15-element boiler, and 10 feet 8¾ inches long in a 16-element boiler, and all are 3 feet 6 inches outside diameter and 1½ inches thick. Each drum has the following exterior fittings: Auxiliary feed, main feed, safety valve, water column, try cocks, manhole (12 × 16), surface blow, stop valve, salinometer valve and steam gauge.

The main and auxiliary feed valves are both 1½ inches combination check and stop valves, and on entering the boiler both feed lines unite, discharging into an internal feed pipe of steel 2 inches in diameter and 7 feet 4 inches long.

The safety valves are Ashton twin spring, each 3 inches in diameter, set for 265 pounds. They may be lifted by hand from the fireroom or from the gun deck.

The water column extends from a point 30 degrees from the top to about 50 degrees from the bottom of the drum, and is fitted with the ordinary gauge and a Klinger reflex gauge. The bottom of the gauge glasses is 9½ inches above the bottom of the drum and 4 inches above the lowest try cock, of which there are four of Navy standard pattern, spaced 5 inches apart vertically.

The surface blow is a 1-inch pipe leading down between the boilers to a Y; where it connects with the bottom blow manifold. This manifold has 1-inch pipes leading to the bottom of each header. The surface blow is fitted with the usual scum pan.

The salinometer pipe is 1 inch in diameter and leads from the head of the drum near the bottom.

The stop valve is 4½ inches diameter and takes steam from a tinned-copper dry pipe 3 inches in diameter, extending nearly the length of the drum. This valve may be closed from the opposite fireroom, but not from the deck above.

Three zinc baskets, each holding five plates 12×6×½, are fitted in each drum. A vertical diaphragm with top curved over to the rear is fitted over each header, its base supported by the diaphragm in the header.

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Each boiler has one furnace and three balanced vertical swinging doors. The ash-pit doors are vertical, inward swinging, and in case of the bursting of a tube the internal pressure will close them. A fire extinguisher with $\frac{3}{4}$-inch supply pipe is fitted in each furnace, worked from the opposite fire-room. The height of the furnace from the dead plate to the lowest row of tubes is 3 feet. The baffling of the product of combustion is effected by means of two diaphragms in each boiler, the lower situated over the eighth row of tubes, and extending from the headers to within $40\frac{1}{2}$ inches of the rear tube supports, the upper situated over the sixteenth row of tubes, and extending from the rear tube supports to within $30\frac{3}{4}$ inches of the headers. These diaphragms are formed simply by pieces of pipe laid in the spandrels of the generating tubes, and not secured in any manner.

The boiler casings are built up, of inner and outer flat sheets of $\frac{1}{16}$ inch thickness, separated by a corrugated sheet of No. 18 U.S.G. in thickness and $\frac{21}{32}$ inch depth of corrugations. These are lined inside with $2\frac{1}{2}$ inches of magnesia covered by one-quarter inch of millboard. The furnaces are lined with fire brick to the height of the fourth row of tubes at the front and the seventh row at the back.

As originally constructed the lining of magnesia and millboard did not extend below the level of the tubes, but on the first dock trial the side casing became red hot abreast the furnaces, and it was decided to continue the magnesia and millboard down to the bottom.

The tube ends are made accessible by means of eight cleaning doors on the front of each boiler, sliding on a horizontal trolley rail at the top, and bolting at the top and bottom. These doors are removable in turn, when moved to the inboard end of the trolley rail, and the end of any desired tube may be uncovered by removing one door and sliding others along the rail.

There are three smoke pipes, each 9 feet 6 inches inside diameter, and extending 119 feet above the grates.

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The boilers were manufactured at the works of the Stirling Company, Barberton, Ohio.

The thrust shaft is fitted with twelve collars, for the horse-shoe thrust block.

The line shafting is in two lengths for each engine, and is supported by two spring bearings.

For each engine there is one section of stern-tube shafting, supported by two stern-tube bearings, and one section of propeller shafting.

SHAFTING.

Each length of shafting is forged solid in one piece, and finished all over, with axial holes drilled from end to end, and also through the crank-pins—these latter used for lubrication.

The crank shafting is in two lengths, each carrying two cranks. A worm wheel for turning the shaft is fitted over the forward coupling.

Total length of shafting, each engine, feet and inches	137- 7¼
Propeller shaft, without casing, diameter in strut bearing, inches	17½
length, feet and inches	25- 8¼
Stern-tube shaft, after stern-tube bearing, feet and inches	4- 4
forward stern-tube bearing, feet and inches	3- 9
length, stern-tube bearing, feet	36

PROPELLERS.

The *propellers* are three-bladed manganese-bronze, inboard turning, the starboard one being left-handed. They are designed as true screws, and the surfaces of the blades are polished and painted.

Number of blades	3
Diameter, feet and inches	17- 3
of hub, feet and inches	4- 6
Length of hub, feet and inches	3- 6
Helicoidal area, square feet	84.5
Disc area, square feet	233.71
Ratio of pitch to diameter	1.058
Pitch (adjustable one foot either way), feet and inches	18–3

MAIN CONDENSER.

These are two, one in each engine room, shell of steel, water-heads of composition, tubes of Admiralty metal not tinned.

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There is a by-pass in the division plate of the waterhead, to allow the circulating pump to discharge directly overboard.

Diameter of shell, feet	7
Thickness of shell, inch	$\frac{5}{8}$
Length over heads, feet and inches	15–10
Tubes, number in each condenser	5,684
diameter, inch	$\frac{5}{8}$
length between tube-sheets, feet and inches	12–10
thickness	No. 18 B.W.G.
space between centers, inch	$\frac{13}{16}$
Cooling surface, each condenser, square feet	11,969

MAIN AIR PUMPS.

Abaft the condenser in each engine room there is one Blake two-cylinder, vertical, double-acting air pump. The valve discs are composed of three sheets of rolled manganese-bronze, each $\frac{1}{32}$-inch thick, and access is had to these valves by means of a manhole suitably placed.

Diameter of cylinders (two) inches	14
pump (two) inches	32
Stroke, inches	21
Diameter of pump valves in bucket, inches	$5\frac{1}{2}$
Number of delivery valves in each end	15
Area through valve openings, square inches	195

MAIN CIRCULATING PUMPS.

For each condenser there is one centrifugal, double-inlet, circulating pump, driven by a compound, vertical, inverted engine. The capacity of this pump is 12,000 gallons per minute at 266 revolutions, and it may draw from the sea, the main drain or the engine-room bilge. The three suction valves are so interlocked that only one can be open at a time.

Diameter of steam cylinder, inches	11 and 22
Stroke, inches	10
Diameter of pump runner, inches	45
Width of pump runner at periphery, inches	4
at hub, inches	$10\frac{9}{16}$
Radius of inlet nozzle, inches	$14\frac{1}{2}$
outlet nozzle, inches	21
Capacity of pump at 266 revolutions, gallons per minute	12,000

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FEED AND FILTER TANKS.

Under each condenser is a combined feed and filter tank of 4,800 gallons capacity. A horizontal partition divides each tank into two compartments, of which the upper is the filter tank, of 1,040 gallons capacity, and the lower the feed tank proper, of 3,760 gallons capacity. Each filter tank is divided into three parts by vertical partitions, and the water flows through all three before reaching the feed tank below. One hundred tons of feed water may be carried in the double bottoms.

FEED PUMPS.

There are two main feed pumps in each engine room. They are Blake, vertical, simplex, center-packed, plunger type, draw water from the hotwell-pump discharge, and discharge into the main feed pipes.

MAIN FEED PUMPS.

Four Blake vertical, simplex, center-packed, plunger type.

Diameter of steam cylinder, inches	14
water cylinder, inches	10
piston rod, inches	2¾
Stroke, inches	12

In each of the six fireroomsis one Blake vertical, simplex, center-packed plunger, auxiliary feed pump, bolted to the center-line bulkhead. Each pump may draw water from the feed tanks, from the hotwell-pump discharge, from the main drain, the auxiliary drain, the sea, the floor of its own compartment, and from the bottom blow pipes of all boilers in its own compartment. It may discharge into the auxiliary feed, into the fire main, or overboard in its own compartment.

AUXILIARY FEED PUMPS.

Six Blake vertical, simplex, center-packed, plunger.

Diameter of steam cylinder, inches	11
water cylinder, inches	8
piston rod, inches	2¼
Stroke, inches	12

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FEED-WATER HEATER.

In each engine room is one feed-water heater, containing 950 square feet of heating surface in 852 tubes. Auxiliary exhaust steam is the heating agent. The hotwell pump delivers water through the heater to the feed-pump suction.

HOTWELL PUMP.

In each engine room is a hotwell pump which may draw water from either air-pump suction pipe, from hose connections on the side of the ship above the water line, and from the main and the reserve feed tanks. It may discharge into the main feed suction line, either directly or through the feed-water heater, or into the main or reserve feed tanks, or auxiliary feed line.

FIRE AND BILGE PUMPS.

In each engine room is one Blake, vertical, simplex, fire and bilge pump 12×10×12. These pumps draw from the sea, the main drain, the auxiliary drain, the drainage manifolds, the engine-room bilge and the crank pits. They discharge to the fire main or overboard.

Diameter of steam cylinder, inches	12
water cylinder, inches	10
piston rod, inches	2¼
Stroke, inches	12

There is placed in the port engine room a distiller circulating pump, Blake, of 350 gallons capacity. It draws water from the sea, and discharges into the flushing system and the distiller circulating pipe, and through the latter into the flushing system or overboard.

AUXILIARY CONDENSERS.

In each engine room there is one auxiliary condenser having 601.4 square feet of cooling surface measured on the outside of the tubes, which are 552 in number, of Admiralty metal, not tinned, 6 feet 10⅛ inches long, ⅝-inch outside diameter, and No. 18 B.W.G. in thickness.

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In the dynamo room is one auxiliary condenser having 1,200 square feet of cooling surface. It has 1,001 tubes similar to the above, but 7 feet 6⅛ inches long.

Each condenser is fitted with a combined air and circulating pump, that for the dynamo condenser taking steam from the bottoms of the dynamo separators. The dimensions of these pumps are, for the engine-room condensers, 18×10×10×12, and for the dynamo-room condenser 12×14×14×12.

FORCED DRAFT.

The closed fireroom system of forced draft is used, there being two Sturtevant blowers in each fireroom.

Number of blowers	12
Type of engine	Two-cylinder, vertical.
Diameter of steam cylinders, inches	6
piston rods, inches	$1\frac{5}{16}$
fan, inches	66
Stroke, inches	5

EVAPORATING PLANT.

There are four evaporators, of Bureau type, each having a capacity of 4,750 gallons per day, or a combined capacity of 19,000 gallons per day. Each evaporator has ninety-six 2-inch brass tubes with a tube-heating surface of 240 square feet. There are three distillers, of Bureau type, each having a capacity of 6,333 gallons per day, or a combined capacity equal to that of the evaporators. Each distiller has 187 tinned-brass tubes with a cooling surface of 127.45 square feet. Circulating water for the distillers is supplied by the distiller circulating pump through the flushing system, or may be drawn from the fire main. The evaporators are situated on the berth deck at the after end of the boiler-room hatch, and the distillers are in a trunk immediately above them.

ELECTRICAL PLANT.

There are two generators of 100 kilowatt and six of 50 kilowatt, all at 125 volts. These are situated in four dynamo rooms, two on each side of the fore-and-aft bulkhead, two at the level of the hold and two above these.

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The 50-kilowatt machines are 6-pole, 400-ampère, designed to run at 400 revolutions per minute while the 100-kilowatt generators are 8-pole, 800-ampère, and run at 350 revolutions per minute.

The generators are of Thresher manufacture, while the engines are built by the Forbes Co.

THE HEATING SYSTEM.

Radiators and heaters throughout the ship are arranged in circuits, each circuit being so connected that it can be operated independently of the others. The circuits are, in turn arranged in groups, each group, having a receiving manifold and a drainage manifold. Steam is taken from the auxiliary steam pipe at the receiving manifold. Here there is a stop valve for each circuit, and a stop valve, an adjustable reducing valve, and a steam gauge at the connection with the auxiliary steam pipe. The drainage manifolds have a stop valve and a check valve for each circuit. Each manifold discharges through a trap, with a by-pass to the feed tanks, the auxiliary condenser or the bilge. By this arrangement each circuit can be entirely isolated, and water from other circuits prevented from backing up into it.

REFRIGERATING PLANT.

An Allen dense-air ice machine of three tons capacity is installed forward on the port side of the berth deck. The refrigerating rooms, four in number, and the ice-making boxes, are adjacent to it. Cooling pipes also lead to the scuttlebutt.

MACHINE SHOP.

This is located on the upper platform between the after fireroom and the engine room, and is divided into two parts by the fore-and-aft watertight bulkhead. Communication between the two is established by a watertight door. The following tools are installed: One screw-cutting back-geared gap lathe, to swing 30 inches over the ways and to take 10 feet between centers; one 14-inch screw-cutting back-geared lathe, to take 4 feet between centers; one column-shaping machine of

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16-inch stroke and 24-inch traverse; one back-geared drilling machine, to drill up to 1½ inches, 14 inches from edge of work; one 16-inch sensitive drill; one No. 1 universal milling machine; one combined hand punch and shears, with 6-inch shear blades, capable of cutting ¾-inch round iron, shearing ⅜-inch steel plate, and punching ⅜-inch holes in ⅜-inch mild-steel plates; one emery grinder with two wheels 12-inch in diameter and 2-inch in face; one 30-inch grindstone; six bench vises. The machine tools are driven by one electric motor in the machine shop.

STEAM PIPING.

The main steam pipes, of seamless-drawn steel, are arranged symmetrically in two systems, one for each side of the ship, each system being independent of the other except for a 9-inch cross-connecting pipe uniting the systems near the main engines. The system for each side of the ship is as follows:

An 8-inch branch pipe leads from each compartment or nest of four boilers. The branches from the forward two compartments unite in a 10½-inch pipe, and this, in turn, unites with the pipe from the after compartment into a 12-inch pipe to the engine room. Stop valves are placed on each branch, and on the main pipe just forward of each junction. The 8-inch branch pipes take steam directly from the auxiliary steam pipe, and so, indirectly, from the boilers. Abaft the engine-room bulkhead the 12-inch pipe has, in the order named, a bleeder connection to the condenser, a 9-inch cross connection to the other side of the ship, a main stop valve, a connection to the auxiliary steam pipe, a separator, an expansion joint, and the main-engine throttle valve.

AUXILIARY STEAM PIPING.

This system consists of two main arteries 8 inches in diameter, extending fore-and-aft through the firerooms, one on each side of the ship, having a pipe 6 inches in diameter, extending from the after firerooms to the engine rooms with branches to the auxiliary machinery, and a 6-inch extension from each

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main artery to the dynamo rooms and the forward-deck machinery, these extensions cross connected in the forward fire-rooms. Each main artery has a 6-inch branch with stop valve for each athwartships pair of boilers. The 6-inch branches subdivide, each into two 4½-inch branches, and each 4½-inch branch connects with a boiler stop valve. The main and auxiliary steam pipes are connected in each engine room.

AUXILIARY EXHAUST PIPING.

An auxiliary exhaust pipe of copper is connected to all of the auxiliary machinery. It has valves to direct the exhaust steam into either auxiliary condenser, either main condenser, either low-pressure receiver, either feed-water heater or into the after escape pipe. The dynamo engines exhaust either into the dynamo condenser or into the auxiliary exhaust pipe.

OFFICIAL TRIALS.

The official preliminary trials of this vessel were held November 21–23, 1905, a full account of which will appear in the next issue of the JOURNAL.

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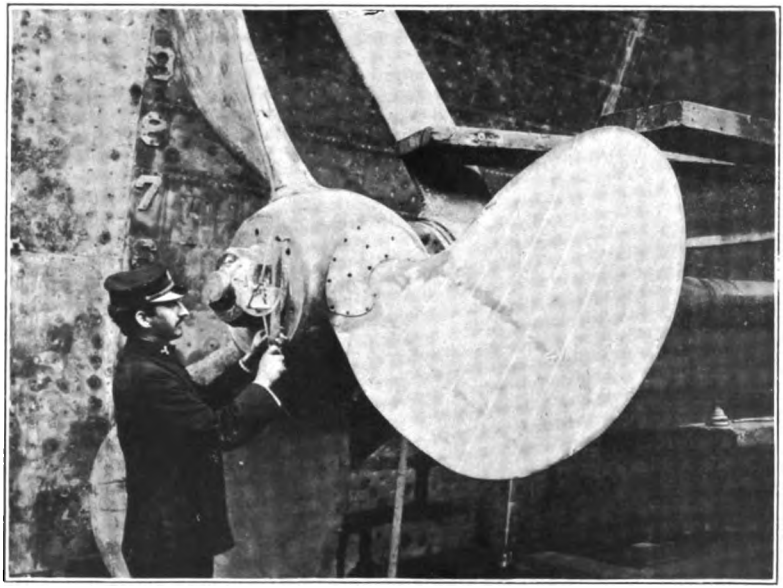

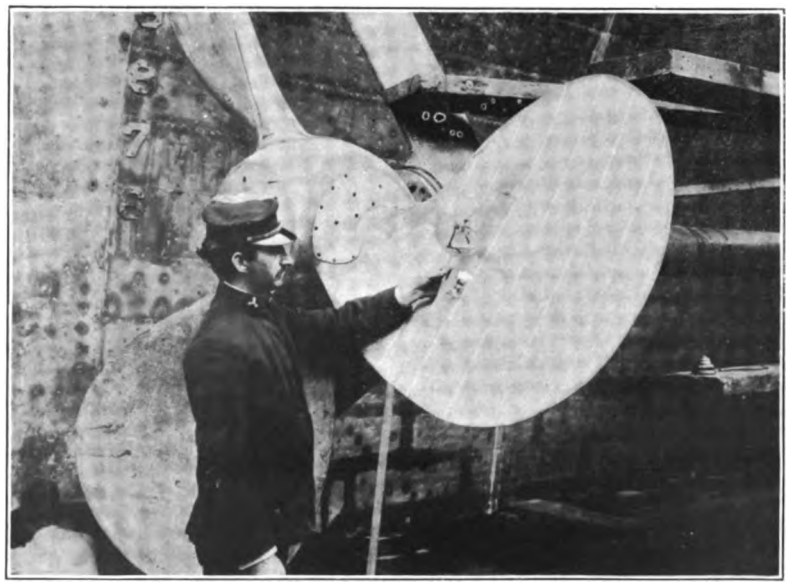

THE WINGATE PITCHOMETER.

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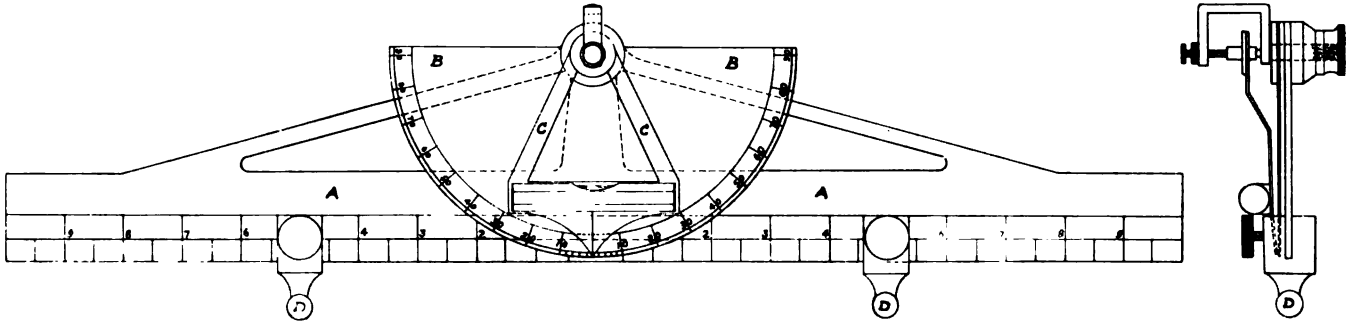

PITCHOMETER
DESIGNED BY C. M. WINGATE,
WARRANT MACHINIST U.S.N

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NOTES.

THE WINGATE PITCHOMETER.

The following notes were furnished by Lieutenant Commander T. W. Kinkaid, U. S. N.

The pitch-measuring device herewith illustrated is the invention of Warrant Machinist Clarence M. Wingate, U. S. N. It can be conveniently used when a vessel is in drydock, but is also available when the propeller to be measured lies on the shop floor.

The pitchometer is based on the formula: Pitch = circumference × tangent of the angle made by the helix with the circumference. As commonly used, the ship being in drydock, with the blade horizontal, the reading taken is the complement of the angle required by the formula. On the back of the blade of the instrument are set forth a column of angular readings and the corresponding pitches for a diameter of one foot. For measurements taken at different diameters it is only necessary to multiply the tabulated pitch by the diameter employed, taken in feet.

When taking readings it is convenient to have parts of helices drawn on the blade with chalk at various radii. The helices are drawn with the aid of a flexible batten through two or three points located on the blade by means of straight edge and square. The instrument is capable of indicating whatever variation of pitch may exist at different points of the helix.

Figure 1 illustrates the construction of the pitchometer. The pendulum index should be plumb when the reading is taken; and to give information as to this fact a spirit level is affixed to the pendulum. The graduated quadrant is pivoted, and may be clamped in any angular position desired. As, in docking, the propeller shaft is seldom level, it is necessary as

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a preliminary step to remove the protecting cap from the propeller nut and apply the two feet of the instrument to the plane surface of the boss, at the same time bringing the quadrant to a zero reading, in which position it is clamped.

THE DÜRR WATER-TUBE BOILER IN H.M.S. *ROXBURGH*.

When the Admiralty decided, on the recommendation of the Water-Tube Boiler Committee, to fit the cruisers of the *Devonshire* class with four different boiler systems, the Dürr boiler was included, partly because of its success in the German Navy and partly as a result of the tests made by the Committee with an experimental installation in the small cruiser *Medusa*. It was therefore fitted in H.M.S. *Roxburgh*. The steam trials of this vessel, completed in May last, gave satisfactory results, and since then the vessel has been put in commission, and considerable interest will be taken in the performance of the steam generators in service.

All the ships of this class have approximately one-fifth cylindrical and four-fifths water-tube boilers; and in the case of the *Roxburgh* the ship, engines and cylindrical boilers were constructed by the London and Glasgow Engineering and Iron Shipbuilding Company, Limited, and the Dürr water-tube boilers were built and fitted on board by the Fairfield Shipbuilding and Engineering Company, Limited, who are the sole licensees for this type of marine boiler in Britain and the Colonies. There are seventeen water-tube boilers of this type fitted in the *Roxburgh*, supplying steam for 16,000 indicated horsepower, the remaining power, 5,000 indicated horsepower, being developed from steam generated in six cylindrical boilers working under Howden's forced-draft system.

The Dürr boilers, to which we more particularly wish to refer, are arranged in three separate compartments, and have a collective heating surface of 41,600 square feet, a superheating surface of 2,245 square feet, and a grate area of 1,085 square feet, with a working pressure of 220 pounds per square inch.

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The construction of the boiler is very similar to the most recent generators of this type fitted in the German Navy. Several modifications, however, have been introduced in the design, resulting from experience gained in the trials of the *Medusa*. The boiler consists essentially of a large steam collector, to which is riveted a vertical water chamber extending across the whole front of the boiler. The tubes, which are of the Field type, with internal circulating tubes, are connected to this water chamber at the front of the boiler, while at the back they simply rest in a lattice-like wall built up of steel flat bars, so that each tube is free to expand independently. Superheating tubes, with similar internal tubes, are also fitted to the upper part of the steam collector, to ensure that absolutely dry steam passes the boiler stop valve. Tube-hole doors are arranged opposite each tube in the water chamber, and also in the steam collector. The water chamber is built of mild-steel plates, welded throughout, and supported by means of screwed stays between the tube holes. It is divided internally by a vertical division plate, through which the stays are also screwed, and to which the inner circulating tubes are connected. The feed water from the steam collector passes through the front part of the water chamber into the inner circulating tubes, leaves the same at the back end, and returns through the boiler tubes to the back part of the water chamber, and from thence it rises to the steam collector.

The two lowest horizontal rows of boiler tubes are 3½ inches in external diameter, and the remaining sixteen rows are 3¼ inches in external diameter. The front end of each is staved up in an hydraulic press, and turned in a lathe slightly conical to make the joint with the steam collector, the axis of the cone being set at a slight angle to the axis of the tube in order to get the required inclination of the tubes to the water chamber. The back ends of the tubes are reduced in diameter, screwed, and closed up by means of a removable forged-bronze cap nut, jointed against a steel collar permanently fixed to the screwed end of the tube. The tube joints in the water chamber are made by forcing the tubes in their coned seat with a small

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hydraulic ram connected to a hand pump, the pressure required being about 2,500 pounds per square inch. With this pressure the tubes are pressed into the tube holes about $\frac{5}{16}$ of an inch, after having been previously bedded, by light blows on a wooden plug, in the cone. The joint surfaces are thinly coated with a graphite composition, which lessens the friction when pressing in the tube, and facilitates removal again later when required. The tubes are readily withdrawn when necessary by means of a screwed rod passed through the tube; and if, by repeated forcing into their seat, the safety collar which is provided at the end of each tube comes so close to the tube plate as to prevent further entry, the tube cone joint may be readily expanded with an ordinary expander, a heavy steel gauge ring, bored to the size of the standard cone, being first placed over the joint surface of the tube.

The inner circulating tubes are $1\frac{3}{4}$ inches in diameter; they are made from thin, folded steel sheets, and held in position at the water-chamber division plate by a bell mouth loosely riveted to the end of the tube to form a hinge joint, which rests on two small clips. The back end of the tube is held in position central with the tube by three springs riveted to the tube.

The two vertical rows of boiler tubes on each side of the boiler are bent right and left in such a way that they form a complete water wall, giving the side casings an effectual protection from the furnace gases.

The superheater tubes connected to the steam collector are $2\frac{3}{4}$ inches in external diameter, with inner tubes $1\frac{1}{2}$ inches in diameter, and are jointed and held in position in a similar manner to that of the water tubes already described. The steam enters the inner tube through slots at the top of the outer division casing, and returns by the outer tube to the inner division casing, the stop-valve orifice being connected thereto.

The gases are baffled on their passage between the tubes by plate baffles resting on the tubes and inserted through the lattice wall at the back of the boiler; the lower row of front

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baffles are held in position by a thin wire rope passed through a row of stays bored hollow for this purpose, and having the end hooked over the nuts of the tube-hole doors below.

The tube-hole doors are of forged steel, circular in form, with joint in the front plate of the water chamber coned in a similar manner to that of the tubes, and, the doors being entered from the inside of the water chamber, the steam pressure tends to keep the cone tight in its seat. The diameter of the door openings is just sufficient to pass the tubes through; and, when placing the tube-hole doors in position, the doors for tubes 3¼ inches in diameter are passed through the larger openings opposite the 3½-inch diameter tubes, and the doors for the latter diameter tubes are entered through a few oval openings. Each door has a screwed end forged solid with it, and a nut, bearing against a light steel cap which covers the outside of the opening, enables the door to be tightened up on the cone joint.

The casings are substantial, yet light in construction, and the large openings through the diamond-shaped lattice work at the back of the boiler, on which the tube ends rest, admits of brushes, or lances for compressed air, being inserted for sweeping the tubes. In front of this lattice work, cover plates, easily portable, and consisting of two thin steel plates with asbestos between, are fitted; these prevent the hot gases passing through the lattice work and ascending in the space between the lattice work and external back end doors of casing.

Air tubes with portable caps, and air doors with pull rods led to the front of the boiler, are provided for admitting a supply of air above and below the fires, respectively, at the back of the furnace.

The water chambers and steam collectors are supported on columns of channel section forming part of the lower front casing, and are also tied to the framework forming the back support for tubes.

Two of the boilers were tested under steam on shore before being fitted on board, at rates of combustion varying from

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18 pounds to 30 pounds of coal per square foot of grate per hour. These trials were very satisfactory, the steam produced being extremely dry.

During the steam trials* of the vessel the boilers worked well; the stoking was carefully and systematically carried out at regular time intervals, clocks in each stokehold being arranged with special dials for this purpose, and from beginning to end of the trials no trouble was experienced with the Dürr boilers.

On the trial at one-fifth power the Dürr boilers evaporated, from and at 212 degrees Fahrenheit, 8.85 pounds of water per pound of coal, the fuel consumption for all purposes being 2.09 pounds per indicated horsepower per hour. On the 30-hours trial at 15,000 indicated horsepower the evaporation was 8.4 pounds per pound of coal, and the mean fuel consumption for all purposes 1.99 pounds per indicated horsepower per hour. On the full-power trial, when the main engines indicated 21,857 indicated horsepower, the Dürr boilers worked with an air pressure of 1 inch, and the cylindrical boilers with 1⅝ inches, the mean evaporation being 7.87 pounds per pound of coal, and the coal consumption 2.3 pounds. The Dürr boilers burned 39.5 pounds of coal per square foot of grate, while the cylindrical boilers consumed 27 pounds per square foot of grate.

—"Engineering."

TRIALS OF 500-B.H.P. DIESEL OIL ENGINE.

Abstract from report of Chief Engineer of British Engine, Boiler and Electric Insurance Co.

The engine was a three-crank inverted vertical, with three single-acting cylinders, numbered 54, 55 and 56, No. 54 being above the idle end of the crank shaft. Each cylinder was 22.05 inches (560 millimeters) diameter, with a piston stroke of 29.53 inches (750 millimeters). The normal speed was 150 revolutions per minute.

The valves were placed in the cylinder covers as usual, and were actuated by levers driven by cams on a horizontal shaft,

* See "Engineering," vol. lxxix, page 681.

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which in turn was driven by a vertical shaft and bevel gear from the idle end of the crank shaft. The cylinders, cylinder covers and exhaust valves were water-jacketed, but the pistons were not. The engine drove a dynamo carried upon a prolongation of the crank shaft.

The air for pulverizing the oil and spraying it into the cylinders was compressed in an independent pair of three-stage vertical air compressors worked by a two-throw crank shaft, belt driven by a motor receiving current from the dynamo upon the engine crank shaft. The air compressors, therefore, though essential to the working of the engine, were not in this case parts of the engine, and in calculating the mechanical efficiency of the engine from the dynamo output and the indicator diagrams this fact should not be lost sight of. Had the compressors been driven directly by the engine, the difference between the work put into the dynamo, which is the brake horsepower, and the indicated horsepower, would have been increased by the power required to compress the air.

The areas of the compressor pistons were:

102.40 sq. in. (660.5 sq. cm.),
32.30 sq. in. (208.1 sq. cm.),
8.79 sq. in. (56.75 sq. cm.),

and the stroke 7.087 in. (180 millimeters), the speed, when compressing to about 64 atmospheres, being about 160 revolutions per minute.

The arrangement, which, of course, is somewhat wasteful, was adopted to meet special conditions. In ordinary cases it is proposed for engines of this size to use Riedler compressors driven from the engine crank shaft by connecting rods or belts.

The dynamo was 12-pole, continuous current, shunt wound, by Lahmeyer & Co., rated to give 450 kilowatts at 550 volts when running at 150 revolutions per minute. The efficiencies given by the makers are—

	kw.	kw.	kw.	kw.	kw.
At	112	225	337	450	562
Efficiency about . .	.88	.925	.935	.94	.935

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and these figures have been adopted in calculating the brake horsepower of the engine corresponding to the measured output of the dynamo.

The power was absorbed by iron wire resistance coils, and the load regulated by appropriate switches. The motor for driving the air compressors was 6-pole, shunt wound, continuous current, by the same makers, rated to give 75 B.H.P. at 630 revolutions per minute.

The calculated efficiencies given by the makers are:

At	Full load.	Three-quarter load.	Half load.	Quarter load.
Efficiency . .	90.5	89	86	76.5m.

DETAILS OF THE FOUR TRIALS MADE.

Four trials were made, the results of which are shown on the accompanying tables.

The first, a preliminary trial, intended to be at full load, but actually a little below; the second at full load, the third at half load, and the fourth with no external load, the engine driving the air compressors only, and, of course, the dynamo and motor which transmitted the power to them.

With respect to the figures in the table, the following explanations should be read:

Line 4.—The diameter of the cylinder of No. 56 engine was gauged. The diameters of the other two were taken from the drawing.

Line 6.—The revolutions were recorded by an engine counter, and the speed indicated by a tachometer.

Line 7.—The water for the jackets was supplied from the town's main, and measured through a water meter which was said to have been recently calibrated.

Line 9.—The discharge pipes from the jackets were conducted to a common pipe discharging into a drain. The same thermometer was used for measuring the temperature of inlet and discharge.

Line 11.—The temperature of the exhaust was measured close to the engine by a mercury thermometer passing through

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a gland in the exhaust pipe, with compressed nitrogen above the mercury to prevent the latter boiling.

	—	Trial I.	Trial II.	Trial III.	Trial IV.
1	Date	13th Feb.	14th Feb.	14th Feb.	14th Feb.
2	Time	3·55 p.m. to 5·55 p.m.	9·17 a.m. to 11·12·15	11·21·15 to 1·30·15	2·1·30 to 2·59·30
3	Duration, minutes	120	115.25	129	58
4	Diameters of cylinders, inches	22.05	22.05	22.05	22.05
5	Stroke of pistons, feet	2.4605	2.4605	2.4605	2.4605
6	Revolutions per minute	150.16	152.8	150.3	150.2
7	Jacket water per minute	166.8	169.85	157.85	140
8	Initial temp. jacket water, °F.	46.3	46.3	46.3	46.3
9	Final temp. jacket water, °F.	125	127.4	104.6	82.1
10	Temperature of outside air, °F.	48	48	48	48
11	Temp. of exhaust gases, °F.	783	806	496	275
12	Analysis of exhaust gases, CO_2	5.6	6.8	3.2	...
	N	42.9	51.3	21.5	...
	Air	51.5	41.9	75.3	...
13	Oil used, pounds	390.3	398	221.1	44.23
14	Oil used per hour, pounds	195.1	207.2	102.8	45.76
15	Blast pressure, atm.	61.5	66.3	50.9	35
16	Maximum pressure shown by indicated diagrams, pounds per square inch	510 515 525	515 525 500	490 480 505	485 480 500
17	M.E.P., pounds per sq. in.:				
	On first piston, No. 54	82.9	80.7	51.6	23.2
	On second piston, No. 55	92.3	93.9	52.6	20.1
	On third piston, No. 56	110	115.6	64.8	33.9
18	Average in the three cyls.	95.05	96.7	56.33	25.4
19	Indicated horsepower	609.3	634.8	363.6	163.3
20	Oil per I.H.P. per hour, pounds.	0.3202	0.3264	0.2828	0.28
21	Output of dynamo, kw.	333	352	168.2	22.24
22	Brake H.P. of engine	475.5	502.5	245	546
23	H.P. absorbed in friction	133.8	132.3	118.6	108.7
24	$\frac{\text{B.H.P.}}{\text{I.HP.}}$	0.78	0.805	0.675	0.334
25	Oil per kilowatt hour, pounds.	0.586	0.5886	0.6113	2.057
26	Oil per brake H.P. per hr., lbs.	0.4103	0.4123	0.4196	0.838
27	Power absorbed by motor, kw.	38	41	31.6	23.6
28	H.P. given out by motor	44.8	48.3	36.8	26.2
29	I.H.P. in compressor cylinders.	36	40	28.8	18.2
30	Power absorbed in belt and compressors, H.P.	8	8	8	8
31	Estimated brake H.P. of engine deducting mean of lines 26 and 27 from line 21, B.H.P.	435.1	458.7	213.8	324
32	Estimate of mechanical efficiency of engine if pump had been driven by it	0.715	0.723	0.588	0.198
33	Oil per brake H.P., pounds	0.444	0.451	0.481	1.415

The above table gives details of the four trials made.

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All these observations ware taken at intervals of ten minutes.

Line 13.—The oil used was from Galicia, costing 100 francs per 1,000 kilos. (£4 per ton), delivered at the works. There is considerable doubt about the calorific value of the oil.

The oil used was very carefully weighed in order to remove any doubts which might be raised by the figures in the heat account of trial III.

Lines 17–19.—The mean effective pressures were calculated from indicator diagrams taken at intervals of fifteen minutes.

It will be seen that the mean pressures in the different cylinders differed considerably.

To afford some check upon the dynamo efficiencies given by the makers, the C^2R losses in the armature and magnet coils and in the shunt regulator were measured. With the full load of 350 kilowatts these were as follows:

	kw.
C^2R loss in armature brushes, etc., ...	8.9
shunt coil, ...	3
shunt regulator resistance, . . .	2.5
Total, ...	14.4

Assuming the iron losses and friction (which could not be measured) to be equal to the above, the total losses would amount to about 28.8 kilowatts, giving an efficiency of 92.4 as against 93.5 claimed by the makers.

Line 22.—The brake horsepower given in this line is at each load the measured output of the dynamo in horsepower divided by the coefficient of efficiency given by the makers for that load. It includes the power absorbed by the motor and air compressor.

Lines 27 and 28 give the kilowatts supplied to the motor which drove the air compressors and the horsepower given out by it.

The difference of 8.8 horsepower between this figure and the figure 44.8 in the first column of line 25 represents the power absorbed by the driving belt and the mechanism of the compressors.

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HEAT ACCOUNTS.

The high percentage of heat unaccounted for on the second trial may be due to imperfect combustion, especially during the earlier part of the trial, when for a short time there was a little smoke from No. 56 cylinder, and the excess of heat accounted for on the third trial is most likely due to overestimation of the weight of the exhaust gases. This weight varies inversely as the percentage of CO_2 in the gases, and any leakage of air into the gas sample reduces this percentage and consequently unduly increases the calculated weight of the gases.

HEAT OIL ACCOUNTS FOR WHOLE TIME PER ONE POUND OF OIL AND PER I.H.P. PER MINUTE.

—	Trial I.		Trial II.		Trial III.		Trial IV.	
	B.T.U.	Per cent.	B.T.U.	Per cent.	B.T.U.	Per cent.	B.T.U.	Per cent.
To calorific value of one pound of oil..	20,050	100	20,050	100	20,050	100	20,050	100
By heat equivalent of work done.....	7,944	39.6	7,794	38.9	8,998	44.9	9,078	45.3
By heat carried off in jacket water...	4,070	20.3	4,110	20.5	5,300	26.4	6,570	32.8
By heat carried off in exhaust gases..	7,030	35.1	6,056	30.2	7,200	35.9	4,402	21.9
By radiation and error	1,006	5	2,090	10.4	1,438	—7.2		
	20,050	100	20,050	100	20,050	100	20,050	100
To calorific value of oil used per I.H.P. per minute	107	100	109	100	94.5	100	93.6	100
By heat equivalent of work done	42.4	39.6	42.4	38.9	42.4	44.9	42.4	45.3
By heat carried off in jacket water...	21.7	20.3	22.3	20.5	25	26.4	30.6	32.8
By heat carried off in exhaust gases..	37.5	35.1	32.9	30.2	33.9	35.9	20.6	21.9
By radiation and error.....	5.4	5	11.4	10.4	—6.8	—7.2		
	107	100	109	100	94.5	100	93.6	100

HEAT ACCOUNTS FOR TRIALS I, II AND III, PER ONE POUND OF OIL, AFTER TEMPERATURE CONDITIONS HAD BECOME NEARLY CONSTANT.

—	Trial I.		Trial II.		Trial III.	
	B.T.U.	Per cent.	B.T.U.	Per cent	B.T.U.	Per cent.
To calorific value of one pound of oil.....	20,050	100	20,050	100	20,050	100
By heat equivalent of work done.....	7,886	39.3	7,893	39.4	8,949	44.7
By heat carried off in jacket water.....	4,440	22.1	4,353	21.8	4,790	23.9
By heat carried off in exhaust gases.....	7,090	35.4	6,100	30.4	6,950	34.6
By radiation and error.....	634	3.2	1,704	8.4	—639	—3.2
	20,050	100	20,050	100	20,050	100

During the two full-load trials the temperature of the exhaust and of the discharge from the jackets continued to rise

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for some time after commencement of each trial, while during the half-load trial both temperatures fell. The inference is that during the earlier parts of the two first trials the temperature of the cylinder walls and pistons was increasing by absorption of heat from the gases and during the third was falling by imparting heat to the gases. To show the effect of these exchanges of heat, heat accounts have been calculated for the periods during which the temperature conditions were fairly constant. The effect upon the oil consumptions and thermal efficiencies was practically negligible.

The distribution of the oil in equal quantities to the cylinders of a multiple-cylinder engine presents difficulties which have not yet been overcome.—"Page's Weekly."

GOLDSCHMIDT'S SYSTEM OF ALUMINO-THERMICS.

(Extract from the Report of the Committee on Science and the Arts of the Franklin Institute.)

The subject of this report has been repeatedly brought to the attention of our members both at stated meetings of the Institute and of its Chemical Section. Scientific and popular accounts, profusely illustrated with experiments, of Dr. Hans Goldschmidt's inventions have been given by members of the Institute as well as by representatives of the inventor, so that it would seem quite superfluous here to enter into a detailed description of the process or to enumerate all the various applications that have been devised or proposed by the inventor.

The sub-committee further deems it advisable to confine its report to "Alumino-Thermics" proper; that is to say, to that part of Dr. Goldschmidt's work which concerns the production and utilization of very high temperatures by means of mixtures of aluminum and with a metallic oxide, such as oxide of iron. Not only have his efforts along these lines led to the most important practical results, but they embrace all that (in the sub-committee's opinion) is novel in the process.

Of the several patents granted to the inventor by the United States Patent Office between March 16, 1897, and August 4,

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1903, three relate to the joining of metal pieces or welding and one to a process of manufacturing homogeneous metal castings.

The heat-producing material known as Thermit, is now manufactured by Dr. Goldschmidt at his Essen works on a considerable scale. It consits of an intimate mixture of finely divided aluminum and oxide of iron (or other metallic oxide), the ingredients being present in chemical proportions—

$$2Al + Fe_2O_3 = 2Fe + Al_2O_3.$$

When this thermit is ignited in one spot, the reaction proceeds with marvellous rapidity through the entire mass without any further supply of external heat. Owing to the high temperature of ignition of thermit, the reaction is started either with a bit of burning magnesium ribbon, or, better, by the aid of a special ignition powder, consisting of peroxide of barium and aluminum powder.

The extraordinary thermal effects of the reacting mass (its temperature is said to approach 5,400 degrees Fahrenheit,) are due to the sudden disengagement of a vast quantity of heat in a relatively small volume of material. The only products are metallic iron and oxide of aluminum, which retain practically the entire heat of the reaction, and, being liquid, separate into two layers—the metal below and the alumina-slag on top.

As already stated, the principle use of thermit is in joining metal pieces, especially of iron and steel.

Thermit welding has been successfully applied during the past four years in a great number of cases, for some purposes on an extensive scale. Conspicuous among these uses of thermit is the welding of street-railway rails at the joints for the construction of continuous track. Many thousands of such welds have been made in England and Germany, and, while the method has not so far found introduction on a large scale in this country, it has been adopted and employed for repairs of the more costly and complex parts of tracks, such as crossings, switches, etc., with excellent results, as one of the members of the sub-committee can attest from personal experience.

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Special mention should be made also of an advantage in thermit welding, in that it dispenses with all bulky equipment, for it requires only a few simple appliances in addition to the thermit itself. Of the latter, the Goldschmidt works now supply "welding portions," each containing the quantity required for one weld. In the case of rail ends, the weld is made by bringing the molten iron resulting from the reduction of the thermit in direct contact with the rails, so that it will adhere to the parts to be joined, and thus unite them.

Another method of thermit welding, applicable to wrought-iron pipes and angle iron, depends entirely upon the heat obtainable from thermit, the reduced iron not being allowed to adhere to the welded parts. To this end it is only necessary to apply the molten thermit mass in such a way that the slag (alumina) first comes in contact with the cool surface of the articles so as to form a protecting layer, and thus prevent the metal which follows from adhering. While the ends of the pieces are brought to the welding temperature, the weld is produced by application of pressure, but without disturbing or changing the section of the welded parts.

Thermit welding has also proved most valuable as a means of repair in break-downs of machinery. Many cases have been reported where otherwise exceedingly difficult, if not impossible repairs were made by its means, and so far as the sub-committee is able to ascertain, with remarkably uniform success. It is to be noted in this connection that the repairs which have been made by the aid of thermit include smaller operations, such as replacing a tooth in a steel wheel, as well as the welding of very large pieces, like the stern posts and crank shafts of trans-Atlantic steamships.

Still another ingenious application of the thermit reaction is in preventing the formation of cavities in large steel castings. A box of specially-prepared, so-called anti-piping thermit, attached to an iron rod, is pushed into the ingot, just as a crust is beginning to form. The heat produced is sufficient to keep the metal liquid where the piping occurs and allows more liquid metal to be poured in to fill the cavity.

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It is the opinion of the sub-committee that the invention of thermit and its application in welding and casting metals constitutes a great step in advance in the production and utilization of high temperatures. While it is recognized that Dr. Goldschmidt was not the first one either to observe or to apply the heat liberated in this reduction of metallic oxides with aluminum, he must be credited with having developed the manufacture of thermit, with the practical application of the enormous heat energy stored in this product, and with extraordinary perseverance and resourcefulness in bringing these inventions to commercial success and thereby creating a new industry.

The chief obstacle to the more extended use of thermit in this country has been the rather high price of the article with the added import duty. It is to be expected, however, that this barrier will be removed as soon as thermit can be supplied by an American factory.

The sub-committee unanimously recommends the award of the Elliott Cresson Medal to Dr. Hans Goldschmidt for his inventions in "Alumino-Thermics," by the Franklin Institute.

THICK FIRES.

It is the prevailing opinion of some owners of power plants that it is necessary, when a boiler is worked at a high rate, to maintain correspondingly heavy fires. It is argued that thin fires are well enough for slow rates of combustion, but as the call for steam increases it must be met by an increased thickness of the bed of coal on the grate. Where heavy fires are carried it is a common thing for the fireman to shovel in all the coal he can conveniently supply, going so far as nearly to fill the opening at the firedoor, leaving little if any room for a future supply until that already in has been pushed back to make room for more. The ordinary fireman is likely to favor this method, for the reason that he can introduce large quantities at a firing, and afterwards he is not obliged to give the fires much attention for perhaps an hour's time, when he will again fill the furnace full in the same manner as before.

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This method of firing with most of the high-class bituminous coals in use in the Eastern States requires from time to time the use of the slice bar for breaking up the bed of coal. Whatever necessity there may be, according to the popular idea, for carrying heavy fires, so far as the amount of work involved is concerned, it is really more laborious for the fireman than it would be to keep the fires comparatively thin and supply small quantities of coal at each firing. An explanation of the favor which this method receives may be that the class of labor generally employed for firing considers the muscular effort required much less of a task than the more frequent and careful attention which is required when the fires are kept at medium thickness.

So far as the engineering merits of thick and thin fires are concerned, the fact is more capacity can be obtained from a boiler when a fire of medium thickness is carried and proper attention is given to its condition than can be realized by any system of management when the fires are exceedingly heavy. Advocates of thick fires, who take the ground that they are a necessity when boilers are forced, are entirely mistaken. As to the economy of the two, some persons maintain that heavy fires give the more economical results, but this is questionable. Valuable information on this point is afforded by two evaporative tests made by Mr. George H. Barrus. They were made on a 72-inch return-tubular boiler having 100 3½-inch tubes 17 feet long. The heating surface amounted to 1,642 square feet and the grate surface to 36 square feet, the ratio of the two being 45.6 to 1. On the thick-fire test the depth of the coal on the grate varied from 8 to 20 inches, being heaviest at the rear end and lightest at the front end. On the thin-fire test the depth was maintained uniformly at about 6 inches. The coal used was New River semi-bituminous. The tests showed that the increased evaporation due to thin fires amounted to 11.5 per cent.

This result, which can probably be duplicated by tests of the same character made by other engineers, shows forcibly the importance of attention to boiler-room management. A

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salesman who told a story of improving the economy of a whole boiler plant ten per cent. would probably be ranked as a gifted prevaricator. The change in conditions of firing effected such a result in this case, and "The Engineering Record" is firmly of the opinion that there are many plants in this country where the fuel bills could be cut down materially by careful attention to such matters. Owners of plants pay large sums for expensive equipment and then leave the most important part of the management, the firing of the coal, to the poorest paid and generally the least careful employés. The boiler room is the most vital part of a steam plant, and its management deserves careful, continuous supervision by somebody who understands the way to get the best service from the fuel.—"Engineering Record."

TURBINE BLADES.

Many experiments have been made into the wear of the blades of steam turbines to determine the cause of this trouble, and the latest to conduct an investigation are the firm of Sulzer Bros. It is undoubtedly a fact that the continued success of the turbine hinges largely upon this one point of blade wear. If the blades of a particular machine should have to be renewed frequently, because of deterioration and consequent loss in economy in the use of steam, that turbine would not make a commercial success. The Sulzer experiments indicate that the wear is due to the particles of water in the steam, and that if water is present in steam that flows at extremely high velocities, as is the case in single-stage turbines, the cutting is apt to be rapid. With superheated steam, or with saturated steam in multi-stage turbines, the cutting is not a serious matter.

The above experiments are in a measure confirmed, thinks an American contemporary, by the experience of those using steam injectors for boiler feeding. This cutting action seldom occurs in the steam nozzle, but frequently does take place in the combining tube where water is flowing at comparatively low velocities. It should be noted, however, that this water

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is mainly the feed water pumped up by the injector, and contains more or less mineral matter, the extent of the cutting probably depending upon the quantity of such foreign particles entrained in the water. In the case of the turbine, however, the water particles suspended in the steam are or should be nothing but pure distilled water, formed by steam condensation after leaving the boiler. Probably no injector would give trouble by cutting if used to pump distilled water, assuming, of course, that the tubes were of a material that would not be corroded by the water. It is thought that no turbine, in which the steam is used at moderate velocity, will be likely to deteriorate rapidly under ordinary operative conditions.—"Page's Weekly."

A STUDY OF THE CORROSION OF CONDENSER TUBES.

By A. Humboldt Sexton.

(Reprinted through the courtesy of the "Engineering Magazine.")

The corrosion of condenser tubes is one of the difficulties which the marine engineer has constantly before his mind, for not only do the failures thus caused give him endless trouble and put him to considerable expense, but the corrosion takes place in so many ways and seems to be so erratic that it is almost impossible to guard against it, and in the minds of many engineers there is a feeling of uncertainty and insecurity which is far from pleasant.

I have had the opportunity of investigating certain cases of bad corrosion of condenser tubes, and have devoted a considerable amount of time to the experimental study of the subject; this paper is therefore submitted as a summary of the present position of our knowledge with reference to this subject, and at the same time as a contribution to its further study. It is not hoped to answer definitely all the questions that may be asked, or to solve all the difficulties that arise in practice, but perhaps to make clear the principles on which corrosion de-

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pends, and to record results which may serve as a basis of future work.

At the outset it may be advisable to say a word or two as to the phenomena of corrosion as seen in condenser tubes, and all the points mentioned will be referred to later. It is assumed that the tubes are of brass, containing 70 per cent. of copper and 30 per cent. of zinc, which is the composition most generally used. When the tubes are removed from a condenser as being worn out, or at any rate as being unfit for further use, probably on account of the failure of some of them, they will usually be in one of three conditions:

(1) The tubes may be worn down nearly uniformly, the thickness of the metal being often considerably reduced, though, of course, it will be thinner in some places than in others. On making a section the remaining portion of the tube appears to be of brass quite unchanged.

(2) The tube may be considerably reduced in thickness, and, as before, to some extent uniformly; but there may be pitting here and there, and the inner surface of the tube seems to have become converted into copper—that is, it is dezincified—and this change in some cases may have gone to a very considerable depth.

(3) The tube may have worn down very irregularly, dezincification being well marked, often in patches, and sometimes extending at various places quite through the metal, giving the appearance of small plugs of copper, which may be either firmly adherent or quite loose; probably there will also be holes corroded through the tube—few or many, as the case may be. These plugs of copper and the holes will usually lie almost in one line, this line being at or near the bottom of the tube, when it is in position in the condenser.

Tubes in the first condition will usually have stood well, and may have been in use many years, those in the second condition may have lasted fairly well, but those in the third will usually have had a short life. The tubes which give trouble to the engineer are often found to be in the third condition.

In addition, there may be corrosion or dezincification to a

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smaller extent from the outside or steam side of the tube, but this is usually due to causes different from those which cause the corrosion from the inside.

The tubes also frequently contain more or less deposit, but this varies so much in appearance that it is impossible to give a general description. Sometimes the tubes are nearly clean inside, and almost free from deposit; sometimes there is a thin skin of black material; whilst in other cases, and often where the tubes have stood badly, there is a heavy deposit, black or brown in color and containing a large amount of ferric hydrate. There are also usually patches, and sometimes a layer of a green deposit which is often very irregularly distributed, and sometimes clusters of white crystalline needles, or nodules of white crystals. Very frequently dezincification is very marked beneath the green deposit. The meaning of these appearances will be discussed later.

The corrosion with which we are concerned takes place in ocean-going steamships, in which the tubes are kept filled with sea water and so serve to condense the steam which surrounds them. The evidence is quite conclusive that sea water is the corroding agent, but its action may be accelerated or retarded by the presence of other materials and by the conditions under which the tube is placed.

Both copper and zinc are readily acted on by sea water under ordinary conditions and even at ordinary temperatures, and much more readily at higher temperatures. Brass, which is an alloy of copper and zinc, often with the addition of a small quantity of tin, is also acted on; but if of suitable composition, much less so than either of its constituents alone.

Sea water contains a considerable quantity of salts in solution, the one in largest quantity being sodium chloride, or common salt; it also contains, in solution, a considerable quantity of oxygen and of carbon dioxide. There is no doubt but that the oxygen in solution is largely responsible for the corrosion, and that the carbon dioxide plays some part in it; but the presence of this latter is by no means essential,

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as corrosion will go on, though at a reduced rate, when carbon dioxide has been completely removed.

The formation of the deposit found in corroded brass tubes, in so far as it is derived from the corrosion of the tube itself, is due to the combined action of the water, salt, oxygen and, usually, carbon dioxide, as is evident from its composition. In one case, the deposit from a corroded tube was found to have the composition—

Copper oxide,	30.85
Copper chloride,	3.2
Zinc oxide,	30.8
Iron oxide,	11.4
Insoluble matter,	3.08
Organic matter, carbon dioxide, water,	20.67
Total,	100

These figures are of importance, for they show that in this case, at least, the deposit is basic chloride and carbonate of copper and zinc. It contains copper and zinc in nearly equal quantities, though in the tube the metals were present in the ratio 7.3, so that it is quite obvious, as indeed will be demonstrated later, that the zinc is more readily removed from the brass than the copper. It should also be noted that there is a considerable amount of oxide of iron (as hydrate), insoluble matter and organic matter, which could not have been derived from the tube itself, but must have been introduced from the outside. This is the case in every sample of deposit from badly corroded condenser tubes which I have examined.

A sample of the deposit formed by the action of clean sea water on copper was found to contain:

Copper carbonate,	29.77
Copper chloride,	8.39
Copper oxide,	59.38
Water,	2.46
	100

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A large number of experiments have been made on the action of sea water on various metals, but more particularly on brass, with special reference to this question of corrosion.

Weighed pieces of metal have been exposed to the action of sea water, in some cases for many months, the water being frequently changed so that the action might not be interfered with by the removal of oxygen and carbon dioxide, which must have happened had the same quantity of sea water been used the whole time. Most of the experiments were made at the ordinary atmospheric temperature, but in some cases the tubes and samples were for some weeks heated to about 80 degrees centigrade during the day and allowed to cool at night. It would occupy too much space and would hardly be of interest to quote results of the experiments in detail, but a summary of the results obtained with brass may be of interest. Taking first of all a brass containing 70 per cent. of copper and 30 per cent. of zinc, which is the usual composition of the brass used for boiler and condenser tubes:

The action of sea water was very marked in every case, and, as might have been expected, was much more rapid when the water was heated. A green deposit was formed, exactly resembling that obtained from corroded condenser tubes. Sometimes this adhered to the metal, but at other times was quite loose. In some cases growths of white crystals of a zinc salt were noticed, exactly similar to those already mentioned as occurring in corroded condenser tubes. The zinc was always dissolved much more rapidly than the copper, but the proportion varies with circumstances. As a rule about five or six times as much zinc was dissolved as copper, whilst in the brass, of course, the ratio was 3.7. The action is more rapid at first; the green deposit which forms on the surface of the metal, especially if it be adherent, seems to some extent to protect it from further action.

It will be noticted that if the metals were attacked in this ratio in the tube from which was taken the deposit, the analysis of which has been given, a considerable quantity of zinc must have been removed in solution or in suspension.

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When the brass contains 66 per cent. or less of copper, the action is very different; the amount of copper removed is small and the deposit is nearly white, consisting almost entirely of zinc salts. In a 66/34 brass tube the ratio of zinc to copper dissolved was 146.1 in one case and 214.1 in another, whilst with 50/50 brass, only a minute trace of copper was removed.

The actions by which a brass tube may be corroded are usually divided into three kinds:

(1) General corrosion, in which the copper and zinc are removed practically together (*i. e.*, in the ratio in which they exist in the alloy) by chemical or electric action, the tube thus diminishing in thickness, but otherwise being unchanged.

(2) Dezincification, in which the zinc is dissoved more rapidly than the copper, and thus a surface layer (or local spots of a material very rich in copper) is left.

(3) Erosion, in which the metal is worn away by the mechanical attrition of the water.

Corrosion of brass tubes always primarily takes the form of dezincification—except, of course, in the case of erosion—the sea water attacking the zinc more readily than the copper, the copper being left in a spongy form, which is very easily worn away. Whenever the copper and zinc seem to be removed together, *i. e.*, when general corrosion takes place, it is always because the spongy copper left by the dezincification has been removed by the mechanical action of the water aided by solid matter which may be in suspension, and this is why tubes which are worn away very uniformly show little sign of dezincification.

The question, however, remains to be answered: "Why is the action so much more rapid in some cases than in others? Why is it that whilst in some cases condenser tubes will last ten years or more, in others they fail in a few months, or occasionally even a few weeks?"

Obviously the fault—if fault there be, or at any rate the reason—must be in one of two places. It must either be due to something in the nature of the tubes themselves, or to the

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conditions under which they have been worked. There is no alternative unless we assume some occult cause to explain the apparently erratic behavior. Each view has its advocates, the former being favored as a rule by engineers who use the tubes, but who are not familiar with the process of manufacture, whilst the latter is the view taken by the manufacturer.

I hold no brief for either side; I have investigated the matter as fully as I have been able, both in the laboratory and by practical examinations of cases of failure, and I am quite familiar with the methods by which the tubes are made and the processes through which they pass before reaching the engineer who will use them.

The tubes are, as already remarked, usually made of brass, containing 70 per cent. of copper and 30 per cent. of zinc, this being the mixture almost invariably specified. It is found to be as durable as any other, there being no advantage whatever to be gained by increasing the percentage of copper. A slight variation up or down has no deleterious influence, and, of course an increase in the quantity of copper is not likely to be injurious. Any careless work on the part of the founder is likely to lead to a greater loss of zinc, and, therefore, to a slightly higher percentage of copper. Many specifications are quite unnecessarily rigorous in the variations in the ratio of the two metals which they make allowable.

The composition of the tubes is always uniform. The composition at the two ends is always the same, and there are no spots, richer or poorer, in copper. This has been proved by many analyses; but quite apart from that, the conditions under which the tubes are made render such variation well nigh impossible, and the idea that there may be spots or patches in a tube differing in composition from the rest, which might set up electrolytic action and thus give rise to local corrosion, whilst believed by many engineers, is not in harmony with facts. The coppery plugs or spots already mentioned have been thought by some engineers to be due to fragments of unalloyed copper left in the mixture. This suggests an impossibility, for the spots are always round, and a fragment of

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copper, if left unalloyed in the casting—which is, however, impossible—would be drawn out into a minute thread in the process of drawing through which the tube goes. These spots or plugs are of quite common occurrence in badly corroded tubes, though they are very often overlooked. The inside of the tube, when taken from the condenser, is usually covered with deposit; the outside is covered with grease; and it is not until these are cleaned away that the spots can be seen. The engineer, as a rule, is not concerned to make a minute examination of the failed tubes; he is anxious to get them away and to get new ones in their place, and hence, unless there is some special question likely to arise, the tubes will not be examined and the spots will not be detected. These spots of copper are certainly due to dezincification, as also is the layer of copper sometimes found inside the tube, and the copper almost always retains a certain amount of zinc. In one case a fragment of a coppery patch from a corroded tube contained 3.7 per cent. of zinc, showing that most of the zinc from the original brass had been dissolved away at that spot, and a specimen of dezincified layer from inside another tube contained 3.84 per cent. of zinc.

It is very doubtful whether the use of specially pure metals is any great advantage, though it is no disadvantage, and it is wisest to err—if at all—on the safe side; but the use of specially pure and, therefore, costly metals, seems to entail increased cost without any corresponding advantage. Nothing in the chemical composition of the brass, if it be approximately 70 and 30, will account for the variations in the durability of the tubes. Indeed, it not infrequently happens that with tubes as far as can be detected in all respects indentical as to composition and make, one set will wear well, whilst another will corrode with extreme rapidity.

The mechanical structure of the metal has sometimes been held to be responsible for corrosion. Brass containing 70 per cent. copper and 30 per cent. zinc is a homogeneous alloy. On cooling, it crystallizes as a whole and does not separate into two constituents as the poorer brasses do. It behaves in

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this respect as if it were a chemical compound, but that it is not a true compound is shown by the fact that the elements are not present in simple atomic ratios, that the composition may be varied within wide limits upwards, and to a small ex-

Fig. 1.—Microsection of Brass Condenser Tube, Showing Coarse Crystallization. (Magnified About Fifty Diameters.)

tent downwards, without altering the structure, and by many of its chemical reactions. It crystallizes as a whole and forms a homogeneous crystalline mass. In the casting the crystals are very large and distinct, but they are broken up during the drawing, and are reformed when the metal is annealed. The actual size of the crystals in condenser tubes varies very much, the difference being probably due to variations in the

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time and temperature of annealing. Figures 1, 2 and 3 are microsections of brass condenser tubes magnified about fifty diameters. It will be seen that the size of the crystals is very different in the three cases. This difference in the size of the crystals may have a profound influence on the physical properties of the metal, but has no influence whatever on its resistance to corrosion.

It has been suggested that the crystals being differently oriented might set up galvanic action, so that the lines of junction of the crystals might become centers of corrosion. There is no evidence whatever in support of this idea; and

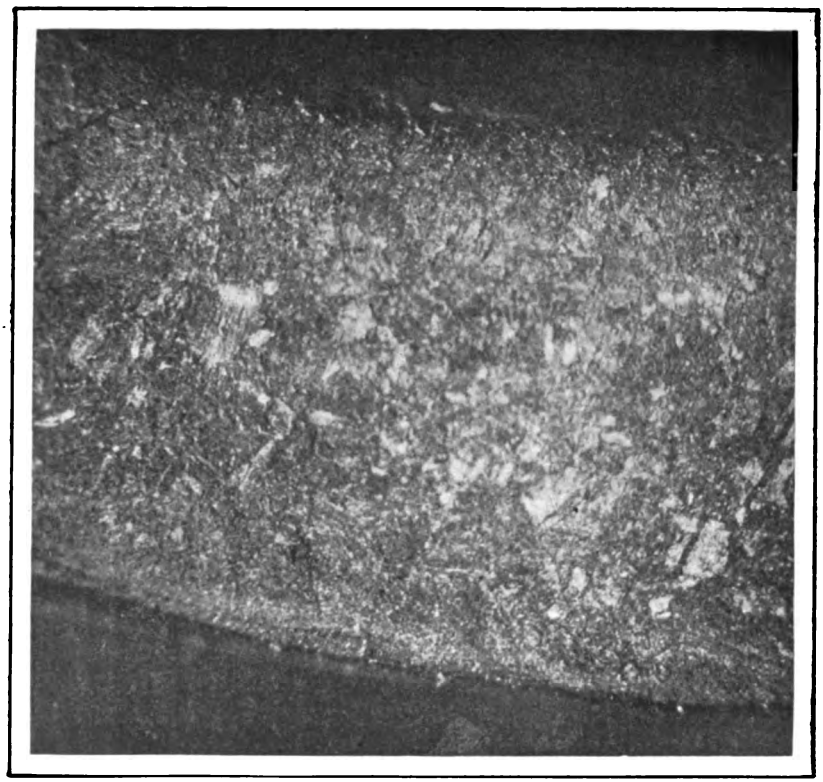

Fig. 2.—Microsection of Brass Condenser Tube, Showing Medium and Rather Irregular Crystallization. (Magnified About Fifty Diameters.)

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even if it were true, it would not account for the formation of pits and corrosion patches a considerable distance apart. I have found that dezincification bears no relation whatever to crystal structure, the line of action as often passing right across a crystal as round its margin.

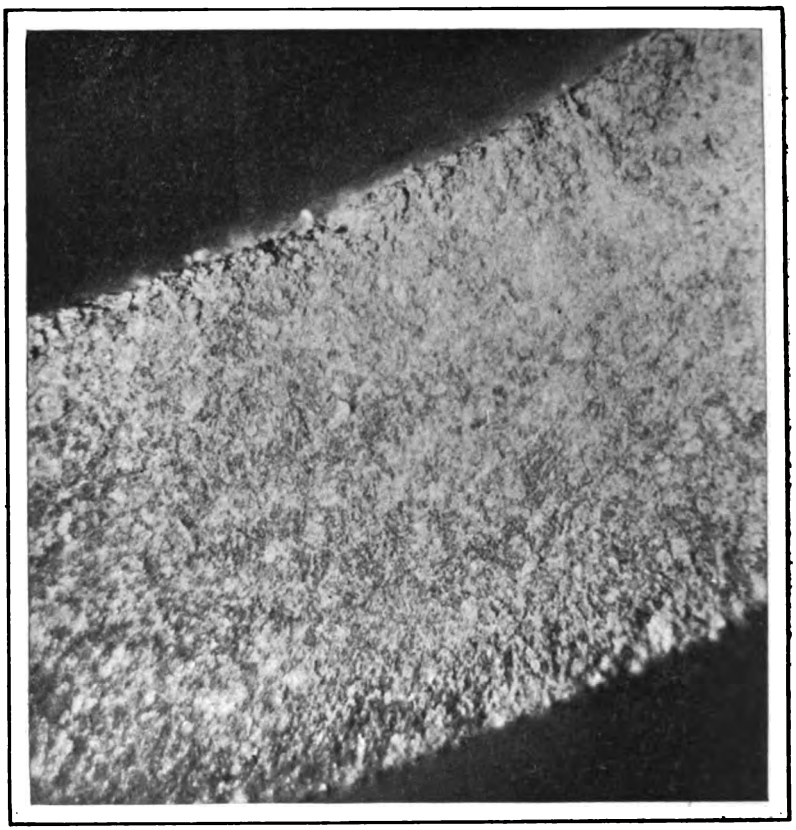

Fig. 3.—Microsection of Finely Crystallized Brass Condenser Tube. (Fifty Diameters.)

It has been contended also that the larger the crystals the more rapid will be the corrosion. This, however, is certainly not the case. I have found that the size of the crystals has no influence whatever on the corrosion of the metal. A very

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large number of corroded tubes have been examined, and it has been found that good and bad tubes—the quality being measured by the amount of corrosion that has taken place—are impartially distributed between those with a coarse, medium and fine structure.

Among methods that have been suggested for increasing the durability of condenser tubes may be mentioned the addition of a small quantity of tin to the alloy. The British Admiralty, for instance, uses an alloy which contains 70 per cent. of copper, 29 per cent. of zinc and 1 per cent. of tin; but it is very uncertain whether this is any material advantage. The coating of the tubes with tin has also been tried, but the protective action, if any, is very small. I am convinced that for this particular purpose nothing is better than 70/30 brass.

I feel quite certain that the cause of variation in the durability of condenser tubes is not to be found in the chemical composition or physical structure of the metal, nor in any variation in the process of manufacture, nor in anything connected with the tubes; indeed, the tube maker, while keeping to the specified composition and passing the tubes through the usual tests for soundness, could not, if he tried, turn out a tube specially liable to corrosion. This is, of course, not the usual opinion of engineers. They say: "Here are two steamers working under exactly similar conditions, and whilst in one the tubes have stood well, in the other they have corroded very rapidly; therefore, the reason must be in the quality of of the tubes." This dilemma may, however, be put in another way. Here are two steamers fitted with exactly similar tubes, selected haphazard out of one large parcel; in the one steamer the tubes have stood well, whilst in the other they corroded rapidly; therefore there must be a difference in the condition of working. The latter is certainly the correct view, for there are so many possible variations in the conditions of working that it is impossible to decide when these are uniform. Two steamers moored at the same berth on a river may be under very different conditions, and so in many other ways there may be differences of condition which, together with differ-

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ences in the way in which the condenser is treated by the engineers in charge, may account for the differences in the corrosion of the tubes.

The solvent action of sea water will account for the slow and steady dezincification which will ultimately wear away the tube; but this will take years, and something more is necessary to explain the irregular corrosion which is so destructive. The corrosion into holes in particular spots seems to indicate some local cause, and at once suggests electrolytic action, the tube itself being the anode; but if this is the case, the difficult thing is to find what can be cathode. It must obviously be some substance strongly electro-negative to the brass.

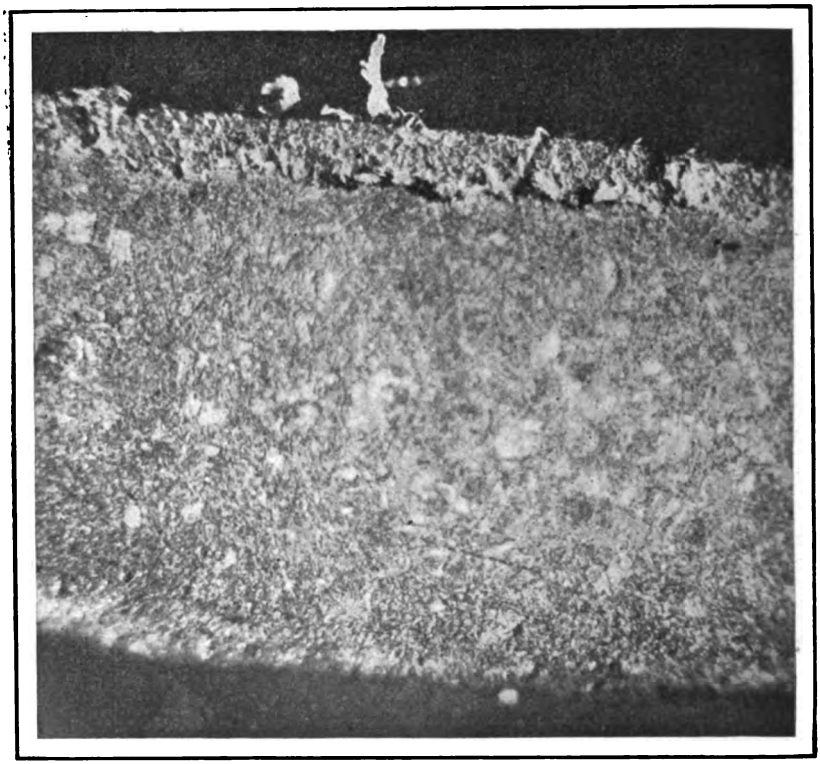

Fig. 4.—Condenser Tube, Showing Layer of Dezincification. (Fifty Diameters.)

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Many experiments have been made as to the influence of the contact of various substances with 70/30 brass—pieces of condenser tubes being used—immersed in salt water, and though the results obtained are not new, they tend to throw some light on the subject under discussion.

Malleable iron and zinc in contact with the brass, as might be expected, completely protect it from corrosion, being strongly electro-positive to it; on the other hand, when a fragment of graphite was placed on the metal, a well marked spot of dezincification (*i. e.*, copper spot) was formed beneath it. The spot was, of course, only superficial, the length of time the experiment was continued not being sufficient to allow the action to go on to any depth. With a piece of coke an exactly similar result was obtained. In both cases the action was more marked when the water was heated. A layer of blacking, which is nearly pure carbon, increased the action of sea water, and also increased the ratio of zinc to copper dissolved. Similar results were obtained with other substances. The green oxychloride of copper seems to protect the copper, and thus to increase the ratio of the zinc dissolved. Ferric hydrate is stated by Professor Tilden to cause rapid corrosion. With a fragment of iron scale, dezincification under the scale was well marked, but copper scale has no such action.

Foul river water has been accused of causing rapid corrosion, and it may be admitted at once that there are many cases of rapid corrosion in the case of steamers plying alternately on such waters and on the sea—perhaps more than on steamers plying only on the deep sea; on the other hand, in many such steamers the tubes last quite the normal time. It will, of course, be admitted that acid waters will be likely to increase corrosion considerably, though 1 per cent. sulphuric acid has little more action than sea water; in any case, the action of an acid water would be expected to be nearly uniform, and, whilst it might accelerate corrosion, would hardly be likely to cause pitting. It is only in very exceptional circumstances, as for instance when acid refuse is discharged into a river, that the water will be sensibly acid.

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Brass tubes exposed for four months to Clyde water—and the Clyde is not regarded as the purest of rivers—taken from the Broomielaw, did not lose seriously in weight, but only became tarnished by the formation of a dark layer, apparently of copper sulphide. Some samples of brass tube were moistened with ammonium sulphide so as to produce dark stains of copper sulphide, and were then put into sea water. The action was at once very marked, the ratio of zinc to copper dissolved being increased to 16.1 per cent. in one case and 12.1 per cent. in another. It would seem, therefore, that a river water containing sulphides might cause a deposit of sulphide which on the tubes being filled again with sea water would set up corrosion, and this being repeated day after day vigorous corrosion might result.

Other suggestions have been made, as, for instance, that the corrosion may be due to electric currents, from the electric lighting or other electric plant on board ship, especially where the single-wire system is used. This may be an occasional cause of corrosion, but I do not think it is at all frequent. It is difficult to see how stray currents can find their way into the condenser, so as to cause electrolytic action. In a paper read before the Institution of Naval Architects on April 3, 1903, Mr. A. W. Stewart describes the corrosion of copper bilge suction pipes on certain steamers of the Russian volunteer fleet, and he comes to the conclusion that in this case electric currents had nothing to do with the corrosion. Corrosion does not seem to be more common in steamers with electric fittings than others, nor does it seem to have increased since the use of electricity became common.

In the above mentioned case, Mr. Stewart ascribes corrosion to the presence of oils and grease. But it is obvious that this is not likely to be the case inside condenser tubes, where grease or oil could not ordinarily find its way. Corrosion on the outside of condenser tubes, which is usually on a small scale, is no doubt often produced by oil or grease carried over by the steam.

I have come to the conclusion that rapid and irregular cor-

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rosion, as distinguished from that due to the normal action of sea water, is almost invariably due to electrolytic action set up by the contact of particles of substances electro-negative to the brass, probably in most cases carbon.

Where can these particles come from? Frequently they may come from outside, especially if the water be drawn from such a position that cinders thrown overboard may be drawn in with it; and cinders have actually been found in corroded condenser tubes. In some cases particles may be derived from the cast-iron doors of the condenser itself, if these are not well protected. A fragment of cast iron would be rapidly oxidized, and for a time would act as a protective agent, but as soon as the iron was dissolved the particles of the residual spongy mass, consisting mostly of graphite, would be electro-negative to the brass and would cause rapid corrosion. If the iron of the door rusted, the graphitic mass would be easily detached, and might be carried into the tube. As already remarked, the corrosion spots usually lie in a line along the bottom of the tube, this indicating that they have been caused by something which settles there. It is also noteworthy that the deposit in corroded tubes nearly always contains a considerable quantity of oxide of iron, which must have been derived from some external source. When a steamer plies on a foul river, the conditions are much more likely to lead to corrosion than on the open sea. On such a river as the Clyde, cinders and fragments of charcoal frequently float on the surface, and may readily be drawn into the condenser. The water may contain sulphides, as evidenced by the very strong odor of sulphuretted hydrogen that may often be detected, and thus patches of sulphide may be formed which may subsequently set up action. Sticky organic and oily matter may frequently be present, and these, though they may have no direct action, may cause the adherence of particles which may in their turn lead to corrosion, especially when coming in contact with sea water. Indeed, the composition of foul river water is so complex and variable that it is difficult to say what action may not take place. The conditions also of such a river vary much with

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the seasons and other circumstances. In time of drought, when the water is low, it will be more strongly charged with impurities than in times of rain, when there is abundance of water. The discharge of acids and other impurities from works is not continuous, but takes place at intervals only, and there may be variations even from hour to hour; and even if the conditions are the same, the state of the tubes as to deposit, and other conditions, may much modify the action.

As to the cure for irregular corrosion, there is none—at any rate after it has made progress; but, like many diseases, if it cannot be cured it can be prevented; and I am strongly of opinion that it is always preventable.

1. The corrosion from the presence of solid particles can take place only if such particles are allowed to rest in the tubes. If the current be strong, therefore, corrosion is little likely to take place, while if it be sluggish, corrosion is very probable. Should a tube become partially stopped for any reason, that tube is specially liable to corrosion. Sluggish circulation is a very common cause of corrosion.

2. The tubes must be frequently cleaned, so that any deposit which is forming may be removed. This is of special importance in steamers running in foul rivers, which may readily pick up substances which may cause the adhesion of objectionable material.

As has been pointed out, tubes that have corroded badly are almost always characterized by the presence of a heavy deposit.

3. The tubes should never be left full of water when the steamer is at rest, but should be run dry and perfectly washed out with clean water, as soon as the day's work is done. This, too, is of special importance in steamers running on foul rivers where objectionable matter may be drawn in, which, during the period of rest, will settle to the botom of the tube and form a lodgment from which it will not be displaced when work is resumed, and so corrosion may be set up, and once started it will go on rapidly under the deposit formed.

In conclusion, I am quite convinced that if the condensers

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be well designed and these precautions be taken, very little will be heard of unduly rapid corrosion; but that unless such precautions are taken the tubes will always be liable to erratic and rapid decay, and the conditions being entirely beyond any power of calculation, no estimate or guarantee can be given as to the probable life of a tube.

Some of the experiments cited are only tentative, and perhaps some of the conclusions may not be fully proved; but I feel quite sure that on the whole they are sound and in the right direction. I will be much obliged if engineers and others who come across cases of corrosion which either bear out or tell against these conclusions will communicate with me and send samples of the corroded tube deposits and anything else that may help to complete the investigation of the causes of the corrosion of condenser tubes and other under-water fittings on board ship.

THE "CONTRAFLO" CONDENSER.

The following notes on a very interesting type of condenser are taken from a description of Messrs. Richardsons, Westgarth & Co.'s Steam Turbine Works, Hartlepool, which appeared in "London Engineering":

The condenser is a very interesting part of the plant, and has been given the name of "Contraflo." This condenser is illustrated in Figs. 1 to 3. The particular condenser here illustrated was constructed for the Hammersmith Corporation Electricity Works, and is for a 1,500-kilowatt steam turbine. The exceptional power of the steam turbine to take advantage of high vacua has led Messrs. Richardsons, Westgarth & Co. to consider how best to attain that end, with the result that they have adopted this type of condenser in their turbine department. The results, they report, are certainly remarkable; but as the efficiency of the apparatus is being exhaustively tested by Professor Weighton at the Durham College, Newcastle-on-Tyne, and as the results will be communicated in a paper to be read by the Professor before one of the engineering institutions, it will suffice at present if we give only an outline

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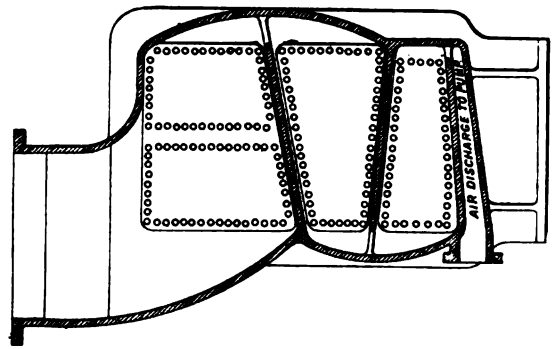

Fig. 3.

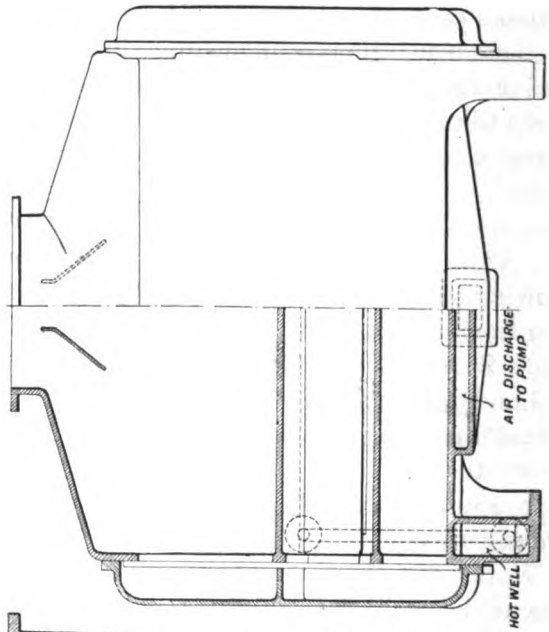

Fig. 2.

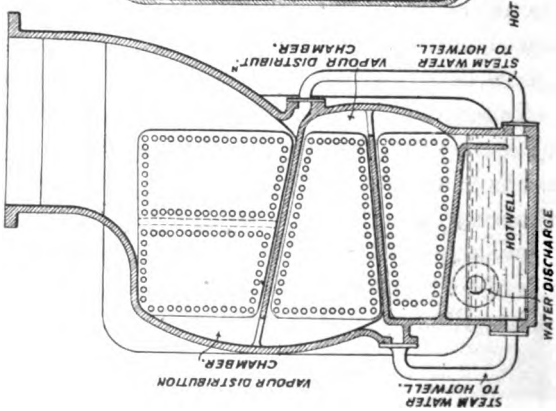

Fig. 1.

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description of the apparatus. It may be added that Messrs. Richardsons, Westgarth & Co. have presented a "Contraflo" condenser, with the necessary plant for its working, to the college, where it will be available for research purposes.

It will be seen by the diagrammatic sectional views, Figs. 1, 2 and 3, that the condenser is divided into communicating compartments of decreasing capacity; and steam and water are drawn off simultaneously from each compartment. Fig. 1 is a cross section through the hotwell end, and Fig. 3 is a cross section through the air-discharge passage. Fig. 2 is a front elevation, half in section. The exhaust steam passes in at the top, flowing through the nests of tubes to the vapor distribution chamber—shown on the left of Fig. 1—common to the two top compartments. The vapor uncondensed passes through the orifice shown in the top diaphragm, dividing the upper two compartments. The water resulting from the condensation that takes place in the upper compartment flows down to the right of the top compartment, owing to the inclination of the dividing diaphragm, and from thence passes through the pipe shown to the hotwell at the bottom of the apparatus. The hotwell is of limited capacity, and the communicating pipes are of sufficiently small dimensions. The vapor uncondensed in the top compartment flows from the lower part of the left-hand vapor distributing chamber (Fig. 1) amongst the tubes in the middle compartment, and a further quantity is there condensed. The water thus produced flows through the pipe on the left of the compartment to the hotwell, any vapor still remaining passing to the upper part of the lower vapor distributing chamber, on the right in Fig. 1, and from thence through the orifice shown, to the lower part of the latter chamber, and finally to amongst the third or lowest nest of tubes, where the condensation is completed, the resultant water flowing directly to the hotwell.

By the gradual elimination of the vapor in the earlier compartments, the air, when it reaches the bottom of the condenser, is comparatively dry, and being drawn by the air pump through a port extending throughout the length of

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the tubes (see Figs. 2 and 3) uniformity of flow is maintained over the entire surface, a temperature resulting which is considerably below that of the water in the hotwell. The combined effect of the rapid trapping and removal of the condensed steam (or steam water), and of the low temperature to which the air is reduced, results in the temperature of the steam water being often well above the temperature corresponding to the vacuum produced. This might be anticipated from the fact that a large part of the water of condensation is drawn immediately to the hotwell from the upper and hotter zones of the condenser, instead of all being carried through to the region of lowest temperature. The important effect obtained is an abnormally high temperature of feed water, and consequently a distinct gain in thermal efficiency. Other practical results are a diminution in the amount of circulating water, and a considerable reduction in the size and weight of the apparatus. The design represents an important advance in surface condensers, and will be of special interest in relation to the high vacuum requirements of steam turbines; whilst decreased weight and space occupied will strongly appeal to the designers of warship machinery. The tests being carried out by Professor Weighton are of a very exhaustive character, and will be of great technical interest.

THE FUTURE OF THE CRUISER.

Considerable significance attaches to the statement that the *Natal*, launched last Saturday, may be among the last armorered "cruisers" floated from private works for the British Navy; there are three of the class still building in the dockyards. It is suggested that in future this type of vessel will be merged with the battleship, because of the great powers of offense and defense required in all large ships of war. This general principle, without any details, and obviously without reference to anything but widely entertained naval opinion, was expressed by a high authority on naval construction—Mr. Albert Vickers—who, it will be remembered, also foretold the

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great increase in the number of high-power guns in battleships, since decided upon in all projected ships, to the exclusion of weapons of intermediate caliber. Our new ship, the *Dreadnought*—the keel of which was officially "laid" this week—will have ten 12-inch guns, the other pieces being only those for defense against torpedo-boat attack. The United States naval authorities have deferred further progress with the design of their new ships until the legislature meets to give permission to increase the displacement in order that a similar arrangement may be adopted. France has decided to enormously augment the gun power of her new battleships, and Germany is seriously considering the question of increasing the depth of water in canals and harbors, so that larger ships may be built to accommodate the modern equipment of guns. The same reasons for increase in ordnance are applicable to armored cruisers; but here there are complications in the problems of design which do not make it so easy to realize the ideal conditions.

The trend for the past two or three years has been towards an increase in the power of the ordnance of warships, and a very considerable impulse has been given to opinion in this direction by the recent naval operations in the Far East. There, under normal conditions, and at long range, the gun failed to penetrate modern cemented thick armor. We are in possession of important details regarding the destruction done to the Russian ships at Port Arthur, since salved by the Japanese Government, and hope soon to have authority to publish these; but it may be said generally that the damage was surprisingly small. In only one spot was the main armor perforated in any of the ships; in both the *Peresviet* and the *Retvisan* there were four holes, and in the *Poltava* one hole, under the armor belt. Few of the shots which perforated the upper deck passed through the armored deck. In the *Peresviet*, which suffered most, four shots out of twelve found their way below the protective deck. These perforations seem to have been largely due to the 11-inch and other guns mounted by the Japanese forces on 203-Meter Hill and other positions around

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Port Arthur, firing at a range of about four miles and at a high angle, so that the projectiles plunged on to and penetrated the decks of the ships, although these weapons were of less power than naval guns of the same caliber. Moreover, Admiral Togo in his actions against the Port Arthur squadron, found it desirable, in order to conserve his relatively small battleship fleet, to fight at long range. It was, no doubt, the consciousness of the ineffective fire at such range—as demonstrated in the Gulf of Pechili—that induced him, when he came to the crucial battle of the Sea of Japan, to refuse to waste ammunition until he got within 6,000 meters, notwithstanding that the Russians opened fire at 9,000 to 10,000, meters. In other words, it was clear to him that even the 12-inch gun is only suited for engagements at about 3 to 3½ miles range. The need, as we have shown, of superior penetrating power of projectiles has been recognized in the case of battleships; and it is equally obvious that if cruisers are to be brought into the line of battle to fight at such range, guns of 6-inch or even 7.5-inch caliber are totally inadequate, and their armor protection may be insufficient. In the subsidiary, or purely cruiser, action in the Sea of Japan the ineffective attack of the 6-inch gun was further established; so that, according to present experience, it seems that the 9.2-inch gun is the absolute minimum for primary armament for large ships. Indeed, there are authorities who consider that the 10-inch or 12-inch breechloader is the best weapon, notwithstanding that its rate of fire is unavoidably less than that of the 9.2-inch piece. But power in each separate shot is the one thing needful.

The question next to be determined is as to whether armored cruisers should be included in the line of battle, or, rather, whether they have a distinctive function. Admiral Togo utilized them in that position, but this was probably of necessity rather than by choice, owing to his deficiency in battleships. There is, however, the significant fact that the practice of our own admirals during the maneuvers of recent years has been to similarly use them. It is felt that without guns of 9.2-inch or higher caliber no warship—as distinct from torpedo craft—

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is of general utility. And this practically leaves no alternative to the naval constructor but to provide so-called cruisers with guns of a caliber almost equal to those of battleships. If the cruiser is not to become abnormally large, the question is as to the number of weapons permissible. Is the cruiser of the future to have high speed at the expense of protection and the number of large-caliber guns? If she is to fight in the line, is heavy armor not almost as essential as powerful guns? But before considering these questions it is necessary to inquire as to whether there are not functions, as, for instance, in the protection of commerce, which demand a higher speed than is essential in a battleship.

This inquiry as to the future of the cruiser raises the whole question of the strategy of the British fleet. We are primarily concerned with our own Navy—an explanation here called for, since strategy and tactics must be based on the relative strength of combatants. A weak power cannot afford to—certainly need not—accept the main strategy of such a strong fleet as that possessed by Britain. It has always been regarded as a first principle that the enemy's coast must be the objective point of our fighting squadron, and that the maintenance of our sea power must be the ultimate aim. So long as our fleet is strong enough to take the offensive against the enemy's ships the command of the sea cannot be said to be in peril. The aim of our fleet, therefore, would be to engage the enemy in squadrons or to contain his ships in their own ports. In such case the cruiser would take her place in the line in fleet actions. It has been popularly regarded as essential that we should have cruisers for protecting our commerce against fast hostile cruisers, but the official view of the present Board of Admiralty is that "the attack or defense of commerce is best effected by concentration of force, and that a dispersion of strength for either of these objects is the strategy of the weak, and cannot materially influence the ultimate results of the war." This view is also taken by Captain Mahan, who, in his "Influence of Sea Power Upon History," says that the harassment and interference with commerce and the at-

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tempt to stop commerce is, when regarded as a primary and fundamental measure, "a most dangerous delusion when presented in the fascinating garb of cheapness to the representatives of the people; especially is it misleading when the nation against whom it has to be directed possesses, as Great Britain did and does, the two requisites of a strong sea power—a widespread, healthy commerce and a powerful navy."

The main strategy laid down by the Admiralty thus involves a concentration of forces; and while our naval authorities do not say that there will not be harassment of our commerce, they refuse to recognize this as a dangerous factor, and do not anticipate that it will necessitate a departure from the main function of our fleet. The percentage of loss in captures by the enemy will be small, partly because our trade routes need not be defined. Even should the enemy's force be dispersed to harass commerce, it will be possible by fleet tactics to counteract the moral influence of such captures, if not to involve the recall of the commerce destroyers. In the last resort it will be possible to detach a superior number of vessels to run the commerce destroyers of the enemy down. The long-entertained idea, therefore, that commerce-protecting cruisers are necessary has been departed from, if it ever found full acceptance, and in any case it need not materially influence the design of vessels which are almost primarily regarded as for fleet engagements. Certainly the opinion is growing that, as Mr. Vickers put it, the type will be merged into the battleship class. This view is entertained not only by many strategists and officials in this country, but also by naval experts in France, America, and even in Germany. But, while it is conceded that the armament of the cruiser should be brought more into line with that of the battleship, there is not the same general agreement regarding the speed and the armor protection of the high-speed battleship which may be "detached," as a last alternative, as the Admiralty put it, to run down the commerce destroyer of the enemy.

In arriving at the compromise between gun, armor and speed, the limiting feature is size, which includes cost. Speed

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is the most costly element of design, for the temptation of the naval designer is to increase the length of hull, as this is of itself conducive to propulsive efficiency. This addition involves a larger target to be protected, and therefore a greater addition to the price of the ship than the augmentation of any other dimension. These problems of compromise are also exercising the minds of naval critics in France, owing to the determination to alter the type of ships to be built, in consequence of the experience in the Far East. But in their case there are factors not obtaining here. It is no disparagement to our friends across the Straits to say that circumstances prevent them spending so much upon their Navy as is possible with our Government, and consequently it is necessary to minimize, as far as is prudent, the expenditure on any one ship. We, on the other hand, are fortunate in being able to place finance in an entirely subsidiary, although nevertheless still an important, position. While, therefore, our Admiralty—and especially the present Board—are certain to give the greatest value for a given expenditure, they do not experience the same monetary limitations as other naval authorities.

Again, the tendency of the weaker power is to depend upon speed, so that in the event of a superior force being encountered, there is the choice of combat and of escape. But no war can be determined without a decisive engagement, and our maintenance of the command of the sea necessitates that our enemy's squadron shall not escape to exercise the discomfiting functions of a "fleet in being." It is therefore easy to understand why many French critics have a leaning towards increasing the speed at the expense of the number of high-caliber guns fitted to the battleship, or the cruiser battleship; while we must still excel in speed as well as in the powers of offense and defense.

Speed was a most important factor in Admiral Togo's success. It is true that his best speed during the battle of the Sea of Japan was 15 knots, but this was quite sufficient to outmaneuver the Russian fleet with their maximum of 12 knots.

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As a consequence, Admiral Togo succeeded, time and again, in heading off the Russian columns, and each time he crossed the head of the lines he so concentrated the fire of all his ships on the leading vessel of each Russian line as to fatally damage, first, the *Osliabia* and the *Souvaroff*, and later the *Alexander III*, which had become leader of the column. Several contributions have recently appeared in the "Moniteur de la Flotte," one of the foremost of the French naval papers, from writers of practical experience and professional ability. They consider the question from the standpoint of the French fleet, and many of them take the view that the future cruiser battleship must sacrifice a few of her large guns in order to increase the speed; but, for reasons of size and cost, they cannot attain to the rate of steaming of present-day cruisers with inferior armament.

M. C. Pierreval makes a strong point, especially from the point of view of the French fleet, when he indicates that high speeds involve greater risks of breakdown, although these may be much minimized by good design and workmanship and by a sufficiently large and experienced staff of engineers. He assumes the meeting of two squadrons, one of 18 knots and the other of 21 knots, differing only by the predominance of speed obtained at the expense of gunpower. In the probable event of the fastest squadron having to run away, he points to the grave risk of a machinery accident. This would involve the sacrifice of the disabled ship; otherwise, the squadron, standing by its "lame duck," would be compelled to fight, notwithstanding inferior guns. He therefore contends that as the gain in speed is more or less uncertain, the preference should be given to the gun. It is computed by one French writer that in a large cruiser an increase in speed of 3 miles per hour would involve an addition of 10,000 horsepower. This, at the minimum computation, would mean 700 tons more weight, which seems rather low. It is equal to 10 per cent. of the defensive power, or to a turret with two 12-inch guns, *i. e.*, 16 per cent. of the offensive power. The problem

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is to determine which of these sacrifices in the fighting qualities is to be made.

In the case of the British fleet we are, fortunately, not so limited in respect of the size of ships as in most other countries, either by the draught available in harbors, by the capacity of our docks, or by the length of our purse. In our new battleship, the *Dreadnought*, we have been able, through the skill of our designers at the Admiralty, to meet all of these requirements. As has been already indicated from semi-official sources, the gun power has not only been enormously increased to ten guns of 12-inch caliber, but in order to add the required increment to the speed, machinery has been adopted which will probably involve an increase of about 500 tons in weight, and, at the same time, the armor protection has been improved. In our new high-speed battleships—the name which must be applied to the armored cruisers of the future—a corresponding general advance will doubtless be realized.

The question is the degree of the advance in each quality. Can the *Dreadnought's* armament be adopted in combination with a higher speed? Two miles per hour more would certainly add to the weight of machinery an equivalent to that involved in at least two 12-inch guns with barbette and mountings. The difference might be met by installing 10-inch guns of 30,000 foot-tons muzzle energy, instead of the 47,000 foot-tons of the 12-inch weapon, or by a combination of 12-inch and 9.2-inch guns. There should be an increase in the speed of our next cruiser battleships. A reduction will, in all probability, be required to be made in the number or caliber of high-power guns to be carried, as compared to the *Dreadnought;* but it is certain that these fast battleships, while equalling the speed of armored cruisers, will be of superior gun power to the existing line-of-battleships. As to whether the changes will be made in connection with the armored ships to be laid down under this year's program remains to be seen.—"Engineering," October 6, 1905.

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THE CLASSIFICATION AND PECULIARITIES OF STEAM TURBINES.

Although the information lately published concerning steam turbines is voluminous, it is difficult to find any unbiased general statements concerning all types of such machine. One reason for this is the fact that few people agree even on such a general subject as the classification of steam turbines, and in view of the different views on this topic it is only natural that equally diverse opinions should be held concerning details. Professor Storm Bull recently read a paper before the Western Society of Engineers in which the classification and peculiarities of turbines were discussed at some length, and this paper deserves attention because the author advocates the adoption of certain German views concerning classification which are more rational than those customarily held in the United States.

The first steam turbine, the 10-H.P. wheel built by C. A. Parsons in 1884, was a reaction machine, while the next type, the de Laval wheel of 1889, was an impulse wheel, using the terms commonly employed. The early machines were at the extremes of turbine possibilities, and development has taken place along the whole line of types between them, which should be classified, Professor Bull says, according to the manner in which the steam acts rather than by their external appearance or speed. The only essential difference between water and steam turbines is that in one the fluid is incompressible while in the other it is compressible. Unfortunately, turbine water wheels are all really reaction wheels, although where all the energy of the water is developed as a jet the wheels are commonly called impulse wheels. This rather unfortunate use of terms has led to the introduction in Germany of new names for these classes, velocity and pressure turbines, which Professor Bull advocates for general adoption. In a velocity turbine, like the de Laval, the steam expands exclusively in passing through the guide wheel or nozzle, and no expansion takes place in the turbine wheel proper, so the pressure is exactly the same on entering the turbine wheel as on leav-

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ing it. This is only possible if the guide and turbine wheels are in separate chambers. In the pressure type, like the Parsons turbine, the pressure gradually decreases from one end of the machine to the other, and the steam expands in the turbine wheels. The classification of steam turbines is not so simple as that of water turbines for another reason, the practicability when using steam of introducing multistage operation, a subject Professor Bull discusses at some length.

With steam of even fairly moderate pressure, the velocity of efflux into vacuum is several thousand feet per second. According to both theoretical and practical hydraulics, the velocity of rotation of the moving blades or buckets should be not far from one-half of that of the steam, in order to get the best efficiency. For steam of 150 pounds pressure by the gauge and a vacuum of 27 inches, the velocity of the steam is not far from 4,000 feet per second; consequently, the velocity of rotation should be nearly 2,000 feet per second. De Laval lets his smaller size turbines run with 30,000 revolutions per minute, and in that manner obtains a velocity of rotation which is not too far away from what it ought to be. For turbines of greater power, the maximum capacity being 300 horsepower, he found it necessary to use a larger wheel, and the limit of the size is determined by the number of revolutions and the strength of the material of which it is made. With these larger sizes the number of revolutions had to be cut down to some 15,000, even though they are made of the very strongest nickel-steel, and the velocity of rotation is only about two-thirds of what it ought to be for the very highest efficiency of the turbine. It has been found impossible to go higher than 300 horsepower, notwithstanding the most careful study of the problem, and de Laval had to invent the flexible and self-adjusting shaft before his turbine could become a practical success.

The purpose of the many-stage or multiple turbine is to get around this difficulty with respect to the high velocity of efflux of steam. Instead of changing the energy of the steam all at once into kinetic energy in a nozzle like that of de Laval, this may be done in several steps, the energy corres-

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ponding to each step being utilized or absorbed by the wheel before the next step is reached. If, for instance, there are fifty steps—as in many of the Parsons turbines—and the difference between the pressure in the boiler and condenser be 150 pounds, then the average drop for each step is only 3 pounds, and the corresponding velocity of the steam comparatively small. However, the impression must not be left that in a many-stage turbine the pressure drop is constant for all the wheels. It may be so, but ordinarily it is only approximately so, and it is not essential to a high efficiency. By this method, then, it is possible to have a very much smaller velocity of the steam, and consequently an equally smaller velocity of rotation, so that the troublesome problem of finding a material which could withstand the centrifugal force generated by the high rotation speed has been gotten rid of.

Returning to the classificution of steam turbines, the question is whether these many-stage turbines are pressure or velocity turbines. The answer is that there are many of both kinds. In a many-stage velocity turbine the pressure in the smaller clearance between the guide wheel and the turbine wheel should be the same as between this latter and the next following guide wheel, and the drop of pressure takes place in the guide wheel. Practically there will be a very small drop of pressure in the turbine wheel, just sufficient to overcome the frictional resistance which the steam has to overcome in passing through the wheel, even though, as is now well established, the drop of pressure does not represent a total loss to the turbine; the real loss being only the lowering of the energy from the pressure on entering the guide wheel to that in the turbine wheel. The best known velocity turbine in the United States is Curtis', in which the steam flows from a series of nozzles into the first turbine wheel, from which it flows through a guide wheel into a second turbine wheel, the pressure remaining nearly constant from the entrance to the first until the exit from the second turbine wheel. From this latter the steam collects in a small reservoir, and is then expanded through nozzles and flows through a turbine wheel, again

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through a guide wheel, to a second turbine wheel and into a second reservoir, and so forth. In this manner there may be three or four pressure stages, each one subdivided in two velocity stages, this latter subdivision serving also to reduce the velocity of rotation.

The Rateau turbine belongs also to the velocity class of turbines. In this there are as many pressure stages as there are turbine wheels, and the space between each turbine wheel and the next following guide wheel forms a reservoir from which the steam flows into the guide wheel, in passing which the steam expands and changes part of its heat into kinetic energy. That this turbine must be a velocity turbine seems evident from the fact that the guide wheels are not provided with openings all the way around, the turbines being what are called "partial wheels." As the steam expands, after passing through successive wheels, the number of the openings or buckets in the guide wheels increases until it becomes necessary to either widen these openings radially or to enlarge the wheels themselves. The important point here is that the expansion of the steam takes place in the guide wheels, except so far as it is necessary to overcome the resistance in the turbine wheels; hence the Rateau turbine is a velocity turbine, but of many pressure stages.

In many-stage pressure turbines the pressure is gradually reduced from the moment the steam enters the first guide wheel until it enters the condenser. The expansion of the steam takes place both in the guide and turbine wheels, the velocity of the steam also increasing, and more rapidly, toward the end than to begin with. The most prominent example of the class of steam turbines is that of Parsons. Depending on the pressure of the steam and of the power of the turbine, the number of wheels in a Parsons turbine may be as large as eighty, although about sixty probably is more nearly the average. It is, of course, quite possible to build a Parsons turbine with a very much smaller number of wheels, and evidently such a turbine would be very much cheaper to build. The drop of pressure would then be correspondingly greater

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for each wheel, also the velocity of the steam as well as that of rotation, and consequently the number of revolutions. If, therefore, it be desired to design a wheel with a comparable low number of revolutions, it is necessary to use a large number of wheels, and presumably this consideration has been the leading one at least in the choice of the number of wheels in the Parsons wheel as it is built. Questions of the strength of materials necessarily also enter the problem, and to a certain extent also the efficiency of the turbine. With a high velocity of the steam the friction per running foot necessarily must be greater than for a low velocity, as it is supposed to vary with the square of the velocity; but at the same time the length of the path along which the steam must flow is being reduced by increasing the pressure drop, so that the question, when the friction is the smallest, is not an easy one to solve.

The various considerations suggested above explain why there is such a great difference in the number of wheels in the principal makes of turbine wheels. Professor Bull believes, however, that the difference will grow less and less as the years pass by and experience accumulates. It must be remembered that the successful steam turbine is but very young in years, and that it will take a number of years before experimenters and designers will agree as to the propositions, and even to the main ideas, in the design which will give the best results. There is room for a very large amount of experimental work of a very intricate nature and on a large scale to determine various problems in the working of the steam turbine, and until that has been done ideas will be more or less confused as to the actual behavior of the steam in the turbine, and also as to the friction of the steam along the vanes and of the friction of the wheels running in steam of various pressures.

Notwithstanding this lack of positive knowledge, it is not to be doubted, Professor Bull says, that the steam turbine is still very far from having reached its greatest state of perfection, and that it has a brilliant future before it. Not all of the designs have been successful by any means; in fact, quite a

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few have been failures. It lies very near to ask the question: How was it possible for the steam turbine, notwithstanding the failure of many designs and of the apparent newness of the idea, to obtain such a marked success? The reasons are of various kinds. First of all, the idea was really an old one, and more work had been done than people in general were aware of. Second, the reciprocating steam engine had apparently reached the highest perfection which it was possible to impart to it, so that inventors had to turn their efforts into a new channel if they were to expect noteworthy results. It is true enough that the efficiency of the reciprocating steam engine has been increased considerably during the last fifteen or twenty years by means of the use of superheated steam, but this gain in economy is shared by the steam turbine as well, so that their relative position is not changed because of the use of superheated steam. A third reason must be sought in the fact that because of its purely rotative motion it avoids those irregularities of motion which are inherent in the reciprocating steam engine, even if it be provided with the heaviest fly wheel. A fourth reason for the rapid introduction of the steam turbine is, no doubt, to be found in the fact that it occupies much less space than its rival, a very important consideration in large cities and on shipboard. A further reason may be found in the fact that it can be regulated much more closely than the steam engine, this because of the very short time—only a very small part of a second—that the steam occupies in passing through the steam turbine, the corresponding time for the reciprocating engine being many times greater.

It will be noticed that an improvement in economy has not been given as one of the reasons for the rapid introduction of the steam turbine, and Prof. Bull is of the opinion that no such improvement has yet been shown. According to the best knowledge possessed, the present status seems to be that for the most favorable load the reciprocating engine still seems to have a slight lead, whereas for a variable load, as is most frequent in electrical lighting and traction plants, the opposite is probably the case. Prof. Bull holds that it is, however,

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almost impossible to express a definite opinion in this respect, as a large part of the material upon which such an opinion must be based is furnished by the manufacturers of the turbines themselves, and which consequently cannot be taken as the very best kind of evidence. It is true that lately this kind of material has been furnished more and more by impartial observers, so that a comparison with the reciprocating steam engine, for which such a large amount of incontrovertible material is available, has become very much easier.

The reasons for the gain in economy, due to the use of superheated steam in reciprocating engines and turbines, are very far from being the same, and an analysis of these will help to furnish an estimate of what can be expected ultimately in this direction from the steam turbine. In the case of the reciprocating steam engine, the gain is due to the reduction of the cylinder condensation, which is due to the difference of the temperature of the saturated steam and of that of the cylinder walls and head, which condensation is greatly intensified by the moisture in the steam adhering to the metallic surfaces. Consequently the gain is not alone due to the fact that the superheated steam is capable of giving up a certain amount of heat without being condensed, but also—and in most cases to a greater extent—because very much less heat is transmitted from the superheated steam to the cylinder than if there has been saturated steam for the same difference of temperature.

In the case of the steam turbine, this difference of temperature is necessarily very much smaller, as no exhaust steam touches the parts which come in contact with the steam on its way through the turbine. On the contrary, if the barrel of the steam turbine is well protected, the temperature of the inside of this barrel can be but very little different from that of the steam at that particular place, hence, the condensation due to this cause is certainly small compared with that in the reciprocating engine. Because of the absence of moisture on the cooler surfaces touched by the steam this condensation will be reduced in the same ratio as for the reciprocating engine; but the difference of temperature being so small, this

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gain, because of the absence of moisture, can only be insignificant. However, even in the ideal case of an adiabatic expansion, condensation is inevitable if saturated steam be used, and such condensation will be prevented if the steam be sufficiently superheated so that its temperature may be lowered without its becoming saturated. In this respect the steam turbine and the reciprocating steam engine are on the same level as to the possible gain due to the use of superheated steam. So far, the analysis has unmistakably shown that the last-named motor has the advantage; but there is an additional reason for the gain in economy by the steam turbine, and that is in the reduced friction which the superheated steam encounters on its way through the turbine as compared with the saturated steam. This friction is entirely negligible in the case of the reciprocating steam engine; but because of the very high velocities utilized in the steam turbine it is a very important factor in its economy. This friction is not alone due to the motion of the steam through the various wheels of the turbine, but also to the rapid revolution of the wheels in this steam. How much the friction amounts to it is at present impossible to say, although some experimental work has been done in this direction, notably by Dr. A. Stodola, of Zurich, Switzerland; but it amounts to a good deal, and there is no question but that it is very much smaller for superheated steam than for saturated.

The use of superheated steam for the reciprocating engine has been accomplished by all kinds of difficulties because of the troubles of lubrication and the expansion of the cylinder due to the high temperature. Although these difficulties have been overcome to a very large extent, yet ordinary practice has limited the superheat to within reasonable limits because of these reasons. In the case of the steam turbine these difficulties do not exist, so far as lubrication is concerned, and but to a very much smaller extent with reference to the expansion of the metal. It is, therefore, possible to use a very much higher degree of superheat for the steam turbine than for the reciprocating steam engine without any bad effects, and this in itself is a distinct advantage.

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Thermodynamics and experience both show that if condensation can be prevented, the efficiency is a function of the differences of temperature between that of the entering steam and that of the condenser. Theoretically, it should be proportional to this difference, but for actual engines it does not increase so rapidly as this would indicate, and there is probably a limit for the higher temperature beyond which the increase would be so small that it would not pay to go, even if circumstances are otherwise favorable. But this limit has certainly not been reached yet, Professor Bull says, and consequently there should be a decided gain for the economy of the steam turbine in the use of more highly superheated steam than has been used so far, or which can be used for the reciprocating engines. Tests of turbines using superheated steam show a gain in economy of about 1 per cent. for every 8 or 9 degrees of superheat.

A steam turbine can, of course, be run noncondensing, and its efficiency, if designed for this purpose, is probably as good at that of the noncondensing engine. But it is a fact established by a large number of tests that it is very much more important to have a good vacuum for a steam turbine than for a reciprocating engine, because the gain in economy is very much greater. It is, therefore, not unusual to hear of 28-inch vacuum for a steam turbine, and such high vacuum has only become possible by the introduction of improved condensers and air pumps, which owe their origin to the demands of the designers of the steam turbines. Why the gain in economy because of a high vacuum should be so much higher for the steam turbine than for the ordinary steam engine, is a question which has troubled a great many persons. The answer is, however, quite simple. Because of the increased condensation it has been found unprofitable to extend the expansion down to the back-pressure line in the steam engine, so that the steam, when exhausted, possesses a considerable amount of energy over and above that at the pressure of the condenser, which energy consequently is lost. In the turbine this is entirely different, as the condensation is

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so small that it does not need to be taken into consideration, and the energy of the steam is changed into a kinetic form, and which, in amount, is proportional to the difference in pressure between that of admission and of condenser. This energy can be utilized, except for friction losses, down to a certain minimum, corresponding to the necessary velocity of the steam when it leaves the last wheel, and which is the same, whether the vacuum is high or low. Although, therefore, there is gain in economy for the steam engine by increasing the vacuum, yet it cannot be so great as for the steam turbine, because the losses of condensation increase so much more rapidly than in the case of the turbine. Professor Bull points out that too much money may be paid for condensing apparatus, water and running expenses in order to get an exceptionally good vacuum; the extra expense may more than outweigh the gain in steam consumption.

It is interesting to note in this connection the success achieved by Rateau in utilizing the exhaust steam from a steam-hoisting engine at a mine for driving a low-pressure steam turbine. As the hoisting engine was only running intermittently, he inserted what he calls a steam accumulator—a mass of iron which absorbs heat during the time when there is an excess of exhaust steam, and gives it off by evaporating water, when there is a deficiency, in this manner enabling the turbine to run continuously. An intermittingly running engine like the hoisting engine can hardly be run condensing, because it takes too long a time to produce the vacuum. But a steam turbine with the accumulator can utilize the exhaust steam with a comparatively high efficiency, as is fully shown by the tests of such turbines in operation. In one test with a pressure of 14.71 pounds absolute the consumption of steam per electrical horsepower per hour was only 39.97 pounds per hour, and for a pressure of 9.37 pounds absolute it was 42.78 pounds.—"Engineering Record."

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SHIPS.

UNITED STATES.

The Navy Department has assigned names to new vessels as follows:

Fleet Collier No. 1, *Erie* is changed to *Vestal.*

Fleet Collier No. 2, *Ontario* is changed to *Prometheus.*

Battleship No. 26, *South Carolina.*

Battleship No. 27, *Michigan.*

Sea-going Tug No. 10, *Patapsco.*

Sea-going Tug No. 11, *Patuxent.*

Submarine Torpedo Boat No. 9, *Octopus.*

Submarine Torpedo Boat No. 10, *Viper.*

Submarine Torpedo Boat No. 11, *Cuttlefish.*

Submarine Torpedo Boat No. 12, *Tarantula.*

Vermont—*Launch of.*—The battleship *Vermont* was successfully launched at the works of the Fore River Shipbuilding Co., Quincy, Mass., on August 31, 1905. This vessel is a sister ship to the *Kansas*, for description of which see page 887, Vol. XVII.

Mississippi—*Launch of.*—This vessel was successfully launched at the works of the Wm. Cramp & Sons' Ship and Engine Building Co., Philadelphia, Pa., on September 30, 1905.

The *Mississippi* is one of two sister ships, the other being the *Idaho*, authorized by Act of Congress, approved March 3, 1903. The contract for the construction of the *Mississippi* was signed January 25, 1904, the price for hull and machinery being $2,999,500.

The ship has a water-line length of 375 feet, a beam of 77 feet, and a normal draught of 24 feet 8 inches, the length over all being 382 feet. The displacement on trial, with a load including 600 tons of coal, is to be 13,000 tons, at which the

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engines, designed to develop 10,000 horsepower, are expected to give a speed of 17 knots. With the full bunker capacity of 1,750 tons of coal and full stores and ammunition on board, the displacement becomes 14,465 tons. The machinery consists of twin-screw, triple-expansion engines, with cylinders measuring respectively 25½, 42 and 69 inches in diameter and a common stroke of 48 inches. These engines are operated by steam furnished by eight Babcock & Wilcox boilers, with a total grate surface of 768 square feet and heating surface of 32,640 square feet, this giving a ratio of 42.5 to 1. The estimated steaming radius at ten knots is 5,750 nautical miles.

The main battery consists of four 12-inch, eight 8-inch and eight 7-inch breechloading rifles and two 18-inch submerged torpedo tubes. The secondary battery consists of twelve 3-inch rapid-fire guns, six 3-pounders, four 1-pounders, two 3-inch field pieces, two machine guns and six automatic guns, caliber .30. The 12-inch guns, arranged in pairs, are in two electrically controlled, balanced, elliptical turrets on the center line, one on the upper deck forward and the other on the main deck aft, with an arc of fire of about 270 degrees and 250 degrees, respectively.

The hull of the *Mississippi* is protected at the water line by a complete belt of armor, nine feet three inches wide and of a uniform thickness of 9 inches for about 244 feet amidships, forward and aft, of which the belt is reduced in width and the thickness is gradually decreased to 4 inches at the stem and stern. Triangular athwartship armor in wake of water-line belt, is 7 inches uniform thickness. There is a complete protective deck extending from stem to stern, the deck being flat amidship, but sloped at the sides throughout and at each end.

Rhode Island—*Official Trials.*—The official preliminary trials of the battleship *Rhode Island* were held over the measured mile off Rockland, Maine, November 1, 1905, and the four-hours' trial in free route November 11, 1905. A detailed description of this vessel and her trials will appear in the next issue of the JOURNAL.

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Louisiana—*Dock Trial.*—A successful dock trial of the machinery of the battleship *Louisiana* was held at the works of the Newport News Shipbuilding and Drydock Company from August 8 to 12, 1905. For a general description of this vessel see Vol. XVI, page 996.

Connecticut—*Dock Trial.*—The main engines of the United States battleship *Connecticut*, which is building at the Navy Yard, New York, were given a dock trial August 21 to 26, 1905. The engines were run separately on alternate days, three days being devoted to the trial of each engine. The actual running time for each engine was about twenty hours. During the first day's trial of each engine the revolutions averaged from 35 to 40, only two boilers being used. On the second day's trial the revolutions were increased to an average of about 60, four boilers being used. On the third and last day's run eight boilers were used, the revolutions averaging between 75 and 80, the maximum being, when the engines did not race, 86 for the port engine and 84 for the starboard one.

Indicator cards were taken at various speeds from each engine. The starboard engine required more horsepower for the same number of revolutions than the port one. This was due to the starboard side of the ship being next to the dock. The port engine showed a tendency to race, and at one time the revolutions reached 104 per minute without any increase in the steam pressure. With the same mean pressure upon the pistons as the cards show for the higher revolutions at the dock, the revolutions should be, in free route, a little over 120 per minute.

Although this was the first time the engines had ever been turned under steam they ran easily and smoothly, apparently being in perfect adjustment. No water was used on any of the bearings, nor allowed to circulate through any of the hollow brasses or guides, with the exception of the thrust bearings, when, owing to the increased thrust due to the ship being fast to the dock, water was allowed to circulate through the cooling coil and to a limited extent through the hollow

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horseshoes. The results of this trial show the excellent character of the work done by the employés at the New York Navy Yard.

The principal dimensions of the *Connecticut* are as follows: Length, 450 feet; beam, 76 feet 10 inches; draught, normal, 24 feet 6 inches; draught, full load, 26 feet 9¼ inches; displacement, normal, 16,000 tons; displacement, full load, 17,666 tons; speed at normal displacement, 18 knots. The engines are designed for a total of 16,500 horsepower at 120 revolutions, with 265 pounds steam pressure at the boilers. They are of the usual vertical, four-cylinder, triple-expansion type, the high and intermediate-pressure cylinders being in the center, and one low-pressure cylinder at each end. The cranks for the forward pair of cylinders and also for the after pair are opposite to each other, while the two pairs are set at right angles. The framing is all of forged steel, consisting of vertical columns well trussed by suitable stays. Piston valves are used on all the cylinders, and are worked by the usual double-bar Stephenson link motion. The crank shafts are hollow, forged in two sections, each section having two cranks opposite to each other. This is done to keep the cylinders as close to each other as possible, and thus reduce the vibration.

The principal dimensions of the engines are as follows:

Diameters of cylinders, inches	32½, 53, and two of 61
Stroke, inches	48
Diameter of piston rods, inches	7¼
Length of connecting rods between centers, inches	96
Diameter at neck, inches	6⅞
crank end, inches	8
of crank pins, inches	17½
Length of crank pins, inches	19
Diameter of crank-shaft bearings, inches	16½
Length of crank-shaft bearings, inches	19½
Cooling surface of each condenser, square feet	10,375

The boilers are of the Babcock & Wilcox type, having 52,752 square feet of heating surface and 1,097 square feet of grate surface, a ratio of 48.1 to 1.

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RESULTS OF TRIAL.

	Port engine.	Starboard engine.
Revolutions per minute	86	83
I.H.P., high-pressure cylinder, pounds	2,260.78	1,942.35
intermediate cylinder, pounds	2,232.11	2,221.42
forward low pressure, pounds	1,035.01	1,124.09
after low pressure, pounds	1,134.49	1,094.89
collective, pounds	6,662.39	6,382.75
Steam pressure, per gauge, pounds	265	255
First receiver pressure, absolute, pounds	98	98
Second receiver pressure, absolute, pounds	35	32
Vacuum, in inches	25.5	25.5
Total indicated horsepower	13,045.14	

—Extract from article by H. P. NORTON, in "Marine Engineering."

ENGLAND.

Natal—*Launch of.*—The armored cruiser *Natal*, launched September 30 by the Duchess of Devonshire from the Naval Construction Works of Messrs. Vickers, Sons & Maxim, Limited, is interesting, not only because this warship illustrates a step in the progress of cruiser design which will ultimately involve their assimilation with battleships, but in her name the *Natal* signalizes the recognition by one more of the colonies of a financial responsibility in the maintenance of the first line of Imperial defence. It is only quite recently that the Parliament of Natal agreed to make a contribution, and for the current year, out of £431,400 paid towards naval expenditure by the colonies, Natal finds £35,000, a fairly large sum per unit of size and wealth; the Cape Colony contributes £50,000; Australasia, £200,000; New Zealand, £40,000; India, £103,400; and Newfoundland, £3,000. The Admiralty's acknowledgement of this generous act by the naming of such a useful cruiser the *Natal*, and the instruction that the launching ceremony should be, as far as convenient, a colonial function, should induce the other members of the Empire to similarly recognize their responsibility in the maintenance of our sea power.

The *Natal*, as we have said, marks the transition stage in the evolution of almost a new condition so far as our warships

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are concerned, and an analysis of her gun power shows the progress made. She mounts, like her three consorts—the *Warrior*, *Achilles* and *Cochrane*, all designed by Sir Philip Watts, K.C.B., Director of Naval Construction—six 9.2-inch and four 7.5-inch guns, as compared with the six 9.2-inch and ten 6-inch weapons in the immediately preceding cruisers for our Navy. She belongs to the first cruiser class in the British Navy, in which the 6-inch gun has been entirely dispensed with, and although this has involved a reduction in the number of guns, the gain is enormous in the penetrating power of each shot fired. The 6-inch gun at its best has a muzzle energy of 6,500 foot-tons; in the most of our ships it does not exceed 5,000 foot-tons. This makes it quite ineffective at the ranges naval tacticians now consider most suitable even in what have hitherto been regarded as essentially cruiser tactics. The 7.5-inch gun, on the other hand, if constructed of 50-calibers in length, as is the case with those in the *Natal*, develops a muzzle energy of over 11,500 foot-tons; and as the possible rapidity of fire per minute is nine shots for the 6-inch gun to six in the 7.5-inch weapon, the advantage, from this point of view, of the lighter weapon does not compensate for its ineffectiveness in power. The 9.2-inch gun, however, is now accepted even in preference to the 7.5-inch piece, and here the 50-caliber weapon may develop a muzzle energy of 24,000 foot-tons. Assuming, therefore, favorable conditions so far as rapidity of fire is concerned, we find that the *Natal* will develop practically 835,000 foot-tons of muzzle energy for each minute's fire, as compared with 829,000 foot-tons in the preceding ships; while but a few years ago our best cruisers, those of the County class, fitted only with 6-inch guns, developed a total of only 542,000 foot-tons. The important point, however, is that the *Natal* may fire per minute 48 shots, which can be effective at 3 miles range, as compared with 24 in the case of the immediately preceding ships—those of the *Duke of Edinburgh* class, eight in the case of the *Drake*, ten in the *Devonshire*, and none in the County class. The following table is instructive:

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FIRE OF PRIMARY ARMAMENT PER MINUTE.

1905. *Natal* (13,550 tons).	24	at 380 pounds =	9,120 pounds and	580,560 foot-tons.
	24	200	4,800	254,400
	48		13,920	834,960
1904. *Duke of Edinburgh* (13,500 tons).	24	380	9,120	441,600
	80	100	8,000	387,200
	104		17,120	828,800
1901. *Drake* (14,100 tons).	8	380	3,040	147,200
	128	100	12,800	619,520
	136		15,840	766,720
1904. *Devonshire* (10,400 tons).	10	200	2,000	101,200
	80	100	8,000	387,200
	90		10,000	488,400
1900. *Kent* (9,800 tons).	112	100	11,200	542,080

Another important improvement in connection with the armament which Sir Philip Watts has introduced in the ships for our Navy is the greater height of the gun platform above the sea level. In the *Natal* and her sister ships all the guns are on the upper-deck level. In gun houses right forward and aft there is located a 9.2-inch weapon, encased in a barbette of 6-inch armor, decreasing to 3-inch at the base. On the four quarters of the citadel there are similar 9.2-inch guns close to the broadside, and, as the forecastle structure is cut away, these guns fire in line with the keel, and have a very considerable arc of fire—180 degrees. Four 7.5-inch guns are placed within casemates on the upper deck, one on each side, near to the center of the length of the ship, the protecting armor here also being 6 inches in thickness. All the guns are loaded and the mountings rotated by hydraulic power. The equipment for dealing with torpedo or submarine attack includes twenty-three quick-firing guns, two 12-pounders, and two Maxim guns. The hoists for supplying all ammunition for these guns are operated by electric motors. In addition, the vessel has two under-water 18-inch broadside torpedo tubes, and one submerged 18-inch torpedo tube firing astern.

The armor is very effectively distributed. There is, in the first place, a complete broadside belt from bow to stern, ex-

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tending from about 5 feet under the water line to the upper deck in the center part of the ship. The thickness of the armor over the machinery spaces, &c., extending over 60 per cent. of the length of the ship, is 6 inches; forward of this it narrows, and is thinned down by two stages, first to 4 inches, and ultimately at the extreme end of the ship to 3 inches in thickness. The protective deck proper extends from the stem to the stern, and ranges in weight from 80 pounds to 30 pounds per square foot. This deck is worked at the lower edge of the armor to assist in the protection of the vital parts of the vessel. Another deck of 40 pounds per square foot forms a crown for the side armor and the armor bulkheads forward and aft, these latter being 6 inches in thickness. The conning tower is 10 inches in thickness, and is provided with an armor tube for the lines of communication to below the protective deck. For communication electric telephones are fitted, and the lighting is also by means of electric power, the double-wire system of distribution being adopted. There are six powerful searchlights. Wireless telegraph equipment will be included.

The dimensions of the ship are as follows: Length between perpendiculars, 480 feet; breadth, extreme, 73 feet 6 inches; displacement at 27 feet draught, 13,550 tons. The normal coal supply carried is to be 1,000 tons, but bunker capacity has been provided for a much increased supply.

The propelling machinery of these cruisers has, as far as possible, been standardized, the general system adopted being that which experience has proved to be most satisfactory for cruisers. The cylinders, all independent castings, are arranged on the Yarrow-Schlick-Tweedy system of balancing, being placed in couples close together, with the valves to the outside; they are supported on an A frame at the rear, with forged-steel columns in front. There are four cylinders arranged for triple compounding, the diameter of the high-pressure cylinders being 43½ inches; of the intermediate, 69 inches; and of the two low pressures, 77 inches. Piston valves are used for both high and intermediate cylinders, and slide valves for the two low-pressure cylinders. The stroke is 42 inches, and

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the total power of 23,500 indicated horsepower is expected to be realized with the machinery running at 135 revolutions. The crank, thrust and propelling shafts are of forged steel, and hollow. The two propellers are of manganese-bronze, with adjustable blades, and work outwards when the vessel is going ahead. The main condensers, oval in section, are of gun metal, with brass tubes, there being two to each set of engines, making four in all, each with its own centrifugal pump of gun metal, with independent engine. These condensers deal also with the steam from the auxiliary machinery of the ship, an arrangement which greatly simplifies the planning of the engine room. The air pumps are entirely separate, and are of Weir's well-known type.

The boiler installation was decided upon before the Admiralty had come to the conclusion that the combination of cylindrical and water-tube boilers had disadvantages which more than made up for any possible economy which, it was assumed, might be derived by the installation of the tank boiler for cruising purposes The *Natal* is fitted with twenty Yarrow boilers and six cylindrical boilers, two-thirds of the power being represented by the former. The proportion of power to weight of machinery is about 10½ horsepower per ton, which is lower than is attainable now with a complete installation of water-tube boilers. The speed anticipated by the new ship is 22.33 knots, and the complement of officers and men is about 700. The *Natal*, it may be added, is in a very advanced stage of construction, the launching weight being 7,400 tons, of which 300 tons is due to the cradle, the practice at the Vickers Works being to defer the launch as long as possible, as there is no difficulty in launching a heavy hull which has been well advanced and clad in armor before being floated. This latter is an economical procedure.—"Engineering," London.

Argyll—*Trials.*—The armored cruiser *Argyll*, belonging to the *Devonshire* class, built and engined by Scotts' Shipbuilding and Engineering Company, of Greenock, has completed her official steam trials with satisfactory results. This cruiser, which has already been described (see JOURNAL A. S.

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N. E., Vol. XVII, p. 430), has a length of 450 feet, a breadth of 68 feet 6 inches, a depth, molded, of 38 feet 6 inches, and at a mean draught of 24 feet 9 inches displaces 10,700 tons. It will be noted that at the trial the mean draught in each case exceeded that designed, but this was due to extra load of water ballast, and is only mentioned here to show that the trial steam performances will be maintained when the vessel has all her war stores on board. On the trial she had all her guns, and was practically ready for service; indeed, the gun trials formed part of the program of her steam tests. The vessel proceeded on her preliminary trial to the South coast, where the ordnance tests were carried out in the English Channel; subsequently she returned to the Clyde to complete the official performances.

The machinery is of the four-cylinder, triple-expansion type, the high-pressure cylinders being 41 inches in diameter, the intermediate cylinders 65½ inches, and the two low-pressure cylinders on each engine 73½ inches, having a stroke of 42 inches. There are sixteen Babcock & Wilcox, and six cylindrical boilers, arranged in four compartments.

STEAM TRIALS OF H. M. S. *ARGYLL*.

Date of trial	Sept. 15, 16, 1905.		Sept. 19, 20, '05.		Sept. 26, 1905.	
Nature of trial	30 *a*		30 *a*		8 *b*	
	Fwd.	Aft.	Fwd.	Aft.	Fwd.	Aft.
Draught of water (mean), feet and inches	24-5	25-7	24-6	25-6	24-6	25-6
Speed, knots	13.9 *c*		20.8 *d*		22.28 *d*	
Steam pressure in boilers, lbs. per sq. in.	180		202		194	
Air pressure in stokeholds, in. of water	Nil		0.6 *e* 0.4 *f*		0.97 *g* 1.05 *h*	
	Stbd.	Port.	Stbd.	Port.	Stbd.	Port.
Vacuum in condensers, inches	25.25	25.5	24.7	24.7	24.5	25.1
Revolutions per minute	85.8	87	125.7	126.6	138.9	138.9
Mean press. in receivers: High, lbs.	110.5	107.8	171	170	172	173
Mean press. in receivers: Inter., lbs.	17.2	16.1	57	53	68.3	67.8
Mean press. in receivers: Low, lbs.	9	11.8	9.2	7.2	15.6	14.3
Mean pressure in cyls.: High, lbs.	35.9	36.9	60.9	62.8	63.3	68.6
Mean pressure in cyls.: Inter., lbs.	12.2	13.5	27.8	28.6	36.76	37.85
Mean pressure in cyls.: Low, fore, lbs.	4.1	4.28	11.7	11.7	16.38	15.75
Mean pressure in cyls.: Low, aft, lbs.	4.39	4.31	12.9	12.3	18.44	17.6
Mean I.H.P.: High	884	920	2,197	2,282	2,522	2,736
Mean I.H.P.: Intermediate	752	841	2,503	2,596	3,652	3,760
Mean I.H.P.: Low, fore	317	334	1,322	1,336	2,047	1,969
Mean I.H.P.: Low, aft	339	339	1,467	1,405	2,305	2,199
Total I.H.P.	2,292	2,434	7,489	7,619	10,526	10,664
Grand total	4,726		15,108		21,190	
Consumption of water (all purposes) per I.H.P. per hour, pounds	20.21		17.28		18.91	
Consumption of coal (all purposes) per I.H.P. per hour, pounds	1.94		1.82		2.2	

a, Hours' coal consumption. *b*, Hours' full power. *c*, (By bearings.) *d*, (Mean of four runs.) *e*, Water-tube. *f*, Cylindrical. *g*, Cylindrical boilers. *h*, Water-tube boilers.

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On the first trial of thirty hours' duration the vessel was required to develop one-fifth of the total horsepower, and it will be noted from the accompanying table that the mean result of the hourly readings was 4,726 indicated horsepower, the consumption of water per indicated horsepower for all purposes being 20.21 pounds. It should be noted in this connection, however, that the proportion of steam used by the auxiliary machinery to the power of the main engines is very considerable. The coal consumption for all purposes was 1.94 pounds. On the second trial, also of thirty hours' duration, the engines developed 15,108 horsepower, and the consumption of water for all purposes was 17.28 pounds, while the fuel consumption was 1.82 pounds. Four runs were made over the measured mile to ascertain the speed, and it was found that this rate worked out to 20.8 knots. On the full-power trial, which took place on Tuesday of this week, the mean draught was 25 feet, and the power developed was 21,139, or 190 indicated horsepower in excess of that guaranteed by the builders. In this connection it should be remembered that the builders of British warships designed at the Admiralty, as in the case of the *Argyll*, are not responsible for speed.

The *Argyll*, on four runs on the measured mile, made a mean speed of 22.38 knots, while the speed anticipated in the design was 22¼ knots; the state of the hull, due to the fact that the vessel had not been in dock for three months, and the problem of propeller design probably accounts for the fact that this vessel has not steamed at quite the same high rate as the other cruisers of the class. Indeed, as we have already pointed out, the Admiralty are accumulating data as to the comparative performances of Admiralty propellers in the vessels of this class, and it may be that the design of the screws of the *Argyll* have not yielded as satisfactory results as those on other vessels of exactly the same design and power. It will be noted that the air pressure in the stokehold on the full-power trial was about 1 inch, and that the consumption of water was 18.91 pounds. The total loss of water during the

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eight hours' trial was 16.8 tons, which is equal to 2.37 tons per 1,000 I.H.P per day of twenty-four hours.

The full results are tabulated in the accompanying table, and an examination of these details will show that they are, as far as the machinery is concerned, most satisfactory. Circling, stopping, starting and gun trials have also been carried out, and the vessel will be at once completed, at the works of her builders, for commission.—"Engineering," London.

FRANCE.

Jules-Michelet—*Launch of.*—The first-class armored cruiser *Jules-Michelet* was launched on the 31st August from the dockyard at Lorient. Her dimensions are as follows: Length, 489 feet; beam 70 feet 2 inches; displacement, 12,750 tons, on a mean draught of 26 feet 3 inches. Protection is afforded by a complete armor belt of hard steel, with a thickness of 5.9 inches in the center, and 3.94 inches at the extremities, extending 7 feet 6 inches below the water-line, over a plate thickness of 0.79 inch. Above the belt the armor will have a thickness of 4.9 inches, to a height of 7.54 feet above the water line. Forward, this armor will be carried as high as the casemates of the 6.48-inch guns. The two armor decks will start respectively from the top and bottom of the armor belt, and will form with it a caisson divided into a number of water-tight compartments. The armament will consist of two 9.4-inch guns in the fore turret, and two 7.6-inch in the after, the turrets being protected by 8-inch armor, with 5-inch hoists; twelve 6.48-inch guns in casemates; and a number of 2.56-inch and 1.85-inch guns; also five torpedo tubes, two of them submerged. Her three engines of triple expansion will develop a total of 29,000 H.P. She will have twenty-eight Du Temple-Guyot small-tube boilers, and four funels, 69 feet in height. Her speed will be about 22.5 knots. When launched, she was already fitted with her propeller shafts and her false keels, and her upper decks were finished. Her launching weight was 5,000 tons, representing 60 per cent. of her construction. She

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was laid down 1st June, 1904, and should be ready for her trials in 1907. Her cost is estimated to be 30,842,731 francs.

In the case of her sister ship, the *Jules Ferry*, some modifications were introduced in the desigus of the boilers, by which the grate surface was augmented and the coal consumption per square foot of grate surface per hour was reduced from 44.64 pounds to 38.3 pounds; in the boilers of the *Jules-Michelet* the proportion of grate surface to heating surface has been still further reduced, and, at the same time, the number of tubes in a boiler has been decreased by one-twentieth, with the result, it is hoped, of reducing the coal consumption to 33.38 pounds per square foot of grate surface per hour.—Exchange.

Submarine Y—*Launch of.*—The new submarine *Y* was launched on the 23d July at Toulon; the order for her construction was given in August, 1901, but she was not actually begun until 22d May, 1902. The *Y* was designed by M. Bertin; she has a displacement of 213 tons, is 140 feet 6 inches long, with a beam of 10 feet 6 inches. She is fitted with explosion motors of 250 H.P., which will be used both for moving on the surface as well as when submerged, and is to have a speed of 11 knots. She is fitted with four torpedo discharges.

GERMANY.

Elsass—*Trials.*—The first-class battleship *Elsass* maintained a mean speed of 18 knots during her six hours' forced-draft trial, the engines making 112.6 revolutions and developing 16,685 I.H.P.

During the measured-mile trials 18.7 knots speed was obtained with 16,812 I.H.P. and 114.9 revolutions per minute.

The coal consumption for a twenty-four hours' trial was found to be 1.73 pounds per I.H.P. per hour, the vessel making 16.5 knots and developing 11,559 I.H.P.

Preussen—*Trials.*—The first-class battleship *Preussen* has recently completed her trials. With the engines developing 18,374 I.H.P., and making 116.7 revolutions, a mean speed of 18.6 knots was attained on the measured mile. With the

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engines developing 9,717 I.H.P., and making 97.7 revolutions, a mean speed of 16.4 knots was maintained. On the twenty-four hours' coal-consumption trial, with the lateral engines developing 6,800 I.H.P., the coal consumption was 1.58 pounds per indicated horsepower per hour.

München—*Trials.*—The third-class cruiser *München* has completed her trials successfully. At her twenty-four hours' full-speed run with the engines making 121.65 revolutions and developing 7,164 I.H.P., a mean speed of 20.4 knots was maintained with a coal consumption of 1.92 pounds per I.H.P. per hour; at a coal-consumption twenty-four hours' trial with the engines making 122 revolutions and developing 7,238 I.H.P., the mean speed was 20.4 knots, and coal consumption 1.92 pounds per I.H.P. per hour. At the full-speed trial on the measured mile, with the engines making 143 revolutions and developing 12,388 I.H.P., the mean speed was 23.45 knots.

Lübeck—*Trials.*—The new third-class cruiser *Lübeck* has been undergoing her trials. She is fitted with turbine engines, but the success attained did not equal expectations, as the speed was not greater than that of her sister ships, the *Hamburg* and *Bremen*, which are fitted with the ordinary reciprocating engines, while the consumption of coal, up to a speed of 16 knots, was greater. The trials were carried out with two small screws, fitted to each of her four propeller shafts, and she is now to go into dock to be fitted with larger screws. In changing from going ahead to going astern, the *Lübeck* was slower than her sister ships with pistons. The engines of the *Lübeck* required considerably less oil, which compensates somewhat for her greater coal consumption at moderate and slow rates of speed. During a twenty-four hours' run, with the engines developing 7,600 I.H.P. and making 670 revolutions, a speed of 20.5 knots was maintained, with a coal consumption of 0.9 kg. (1.98 pounds) per I.H.P. per hour, with the engines developing 9,500 I.H.P., a speed of 21.5 knots was maintained, with a coal consumption of 1,075 gr. (2.149 pounds) per I.H.P. The weight of the turbine engines is 77.5 tons less than the reciprocating engines of her sister ships, and

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the ship seems to be steadier in a seaway. It is interesting to note that in the British third-class cruiser *Amethyst*, which is fitted with turbines, the economy of fuel was progressive from 14 knots up to full power, when she completely distanced her sister ships, the *Diamond* and *Topaze*, both in speed, power and economy of fuel.—Exchange.

Berlin—*Trial of.*—The third-class cruiser *Berlin*, during a twenty-four hours' trial, maintained a mean speed of 21 knots. The engines developed 7,121 I.H.P. and averaged 115 revolutions per minute.

A mean speed of 22.6 knots was obtained during the six-hours' forced-draft trial, the corresponding I.H.P. being 10,857, with the engines averaging 127.6 revolutions.

A speed of 23.26 knots was reached on four runs over the measured mile.

Hannover.—The following are the particulars of the German battleship *Hannover*, recently launched at Wilhelmshaven: Length over all, 430.289 feet; between perpendiculars, 398.628 feet; beam, 72.83 feet; draught, 25.26 feet; engines, triple-expansion, of 16,000 horsepower; boilers, Schultz-Thornycroft, to the number of twelve; speed, 18 knots; armament, four 11-inch guns in two turrets with 11-inch protection; fourteen 6.7-inch guns, ten in the central casemate and four in four separate casemates; twenty-two 3.46-inch and fourteen 1.45-inch guns, four mitrailleuses, and six torpedo tubes, submerged; thickness of armor belt amidships, 9.45 inches decreasing to 5.9 inches, and then to 3.93 inches fore and aft; thickness of armor above the belt, 8 inches; coal, ordinary load, 700 tons; special load, 1,800 tons.—"Engineering," London.

Danzig—*Launch of.*—The third-class cruiser *Ersatz Alexandrine* was launched on September 23d from the Imperial Dockyard at Danzig, and received the name of *Danzig*. Her dimensions are as follows: Length, 340 feet 6 inches; beam, 43 feet 4 inches; displacement, 3,250 tons, on a draught of 16 feet 5 inches. There will be a 2-inch armored deck, and the conning tower will also be protected by 3-inch armor. The

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armament will consist of ten 4.1-inch Q.F. guns, ten 1.45-inch Q.F. guns, with four machine guns and two submerged torpedo tubes forward for 18-inch torpedoes. The engines are to develop 10,000 I.H.P., giving a speed of 22 knots, while the coal capacity is 800 tons, giving a radius of action of 5,000 miles at economical speed.

ITALY.

Napoli—*Launch of.*—The new Italian battleship *Napoli* was launched on September 10th, at Castellamare. The *Napoli* will form a division of four homogeneous ships with the *Vittorio Emanuele*, *Reginia Elena* and *Roma*, but this last will not be launched until this time next year. The *Napoli* was laid down on October 21, 1903. Her length over all is 474 feet; between perpendiculars, 435 feet; beam, 82 feet; draught, 25.8 feet; displacement, 12,624 tons; number of engines, two; number of cylinders in each engine, four; number of boilers (Babcock & Wilcox), twenty-two; engines, 15,200 horsepower under natural draft, and 19,000 horsepower under forced draft; speed, probably 22.5 knots; ordinary load of coal, 1,000 tons; special load, 2,000 tons; radius of action at 12 knots with 2,000 tons of coal, 10,000 miles; armament, two 12-inch, twelve 8-inch, sixteen 3-inch, two 2.95-inch and ten 1.85-inch guns, two mitrailleuses, and four torpedo tubes. Her armor belt will run her whole length and be 9.18 feet broad, with a maximum thickness of 9.84 inches. Protection for 12-inch and 8-inch guns, 7.87 inches; for small guns, 3.15 inches; for conning tower, 9.84 inches. Her complement will be 36 officers and 679 men, or, when serving as flagship, 44 officers and 760 men.—"Engineering," London.

Submarines.—The Italian submarine *Glauco* has just been launched at Venice. This submarine, which is of a new type, has been built in succession to the *Delfino* and the *Triton*, which have not developed altogether satisfactory results. The *Glauco* will have a surface speed of 14 knots, and her radius of action will be 2,000 miles. Four other submarines are in

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course of construction for the Italian Navy; they will be named the *Squalo*, the *Narvalo*, the *Ottaria*, and the *Tricheco*. —"Engineering," September 8, 1905.

JAPAN.

Mikasa—*Accident to.*—On September 11th, a fire broke out on the battleship *Mikasa;* it spread rapidly, reaching the after magazine, which exploded, causing the vessel to sink in shallow water. The loss of life was heavy, amounting to nearly 600, amongst whom were many from other ships who had gone to the rescue.

The cause of the fire is not known nor has the extent of the damage been determined, though it is known that a hole was blown in the ship's port side.

The *Mikasa* is a first-class battleship, having a displacement of 15,200 tons, and speed of 18 knots.

She was the flagship of Admiral Togo during the whole of the Japanese-Russian war.

SPAIN.

Cardenal Cisneros—*Loss of.*—The cruiser *Cardenal Cisneros* sunk on October 30 at 6 o'clock off Muros, on the Galician coast. She was steaming to Ferrol and struck on a rock, owing to the fog. Her commander and all the officers and crew of 540 men were rescued, but the ship could not be refloated. The *Cardenal Cisneros* was an armored cruiser of 6,889 tons and 15,000 horsepower. She was built in 1902 at Ferrol, Spain, was about 364 feet long, had 60¼ feet beam, drew 25 feet of water and cost $3,000,000. The cruiser had 8 inches of armor and her armament consisted of two 9.4-inch guns, eight 5.5-inch guns and a number of smaller guns.

SWEDEN.

Magne—*Trials.*—The torpedo-boat destroyer *Magne*, built by Messrs. John I. Thornycroft & Co., for the Swedish Government, has recently completed her trials and has been sent to Sweden.

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The vessel, which has a length of 219 feet 9 inches, with a beam of 20 feet 9 inches, was designed to attain a speed of 30.5 knots on a three-hours' trial when carrying a load of 50 tons, and for this purpose was fitted with two sets of four-cylinder, triple-compound, condensing engines, designed to develop 7,500 indicated horsepower, steam being supplied by four boilers of the Thornycroft-Schulz type. Generally, the vessel and the machinery correspond to the Japanese destroyers *Shirakumo* and *Asashio*, which were built by Messrs. Thornycroft, and proved, during the recent war, most satisfactory vessels, even in a very heavy sea. The full-power trial of the *Magne* was carried out on August 31st; under the required conditions of load, the vessel attained a speed during the three-hours' test of 30.705 knots, with the engines running at 378 revolutions per minute, and indicating 7,700 indicated horsepower.

MISCELLANEOUS.

Iris.—The following extracts are from a description of the motor launch *Iris*, in "Engineering," London, the various illustrations having been omitted. The *Iris* was designed for a sturdy type of sea-going launch, and built in accordance with the rules of the Marine Motor Association. The hull is built on the plain carvel system, and copper fastenings are used throughout. The following are the principal dimensions of the boat:

Length over all, 30 feet; length, water line, 29 feet; beam, extreme, 6 feet; beam, water line, 5 feet 8 inches; depth amidships, 3 feet 8 inches; freeboard, forward, 3 feet; freeboard, amidships, 2 feet 1 inch; freeboard, aft, 2 feet 2 inches; draught, extreme (at skey), 2 feet; displacement with crew, 2 tons; area, midship section, 4 square feet; block coefficient, 0.451; prismatic coefficient, 0.620; brake horsepower, 24; revolutions per minute, 800.

Speed being a secondary consideration, a very stanch craft was produced. Sea-going qualities and handiness being necessary qualifications, the midship section was kept very large; this, with a wide afterbody, has made her a very handy and

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stable sea boat. The accommodation is large, seating ten persons aft quite comfortably, and providing ample storage space in the fore compartment, which is 6 feet long. The hull is of pine planking $\frac{5}{8}$ inch thick, with teak top strake. The stern dead woods and knees are of English oak cut from grown crooks. Running the whole length of the boat is an American elm keel inside and an Oregon pine keel outside, which forms the principal girder of the boat's frame, and is carried from the stem to within 5 feet of the stern. This keel is 3 inches thick throughout, has a depth of 15 inches at after end, tapering to the scarph of the stem, and secured to the inner hog keel by metal bolts through all. This makes the boat exceedingly rigid and enables her to take the hard ground without damaging the propeller or rudder; at the same time it adds considerably to her sea-going qualities.

The gunwales are in one length, each $2\frac{1}{2}$ inches by 2 inches, of Oregon pine, and are covered with a teak capping, running the whole length of the boat, and below this is a heavy Oregon pine rubbing strake. The motor and reversing gear being mounted on such a rigid channel frame, shallow motor bearers can be fitted, and these are 3 inches wide by 17 feet long, of selected red pine. They are bedded over six 3-inch by 2-inch oak floors, which fit the skin of the boat and extend well up the turn of the bilge. The stern tube is of gun metal, 5 feet long, and has a stuffing gland inside.

Fuel is supplied to the carburetter by gravity from a 10-gallon tank in the fore compartment under the turtle deck. There is a second fuel tank of 21 gallons capacity athwartships under the main thwart, and the fuel contained in this can be pumped into the forward, or running, tank by a semi-rotary hand pump placed conveniently close to the driver. The motor is encased in a neat mahogany housing with hinged tops, and copper trays are fitted under all the mechanism. A canvas spray hood is also fitted for heavy weather, and extends from the coaming at the aft end of the turtle deck to the steering column.

The *Iris* was engined by Messrs. Legros & Knowles, Limited,

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of Willesden Junction, London, N. W. The engine is a four-cylinder petrol motor of $4\frac{3}{4}$-inches bore by $4\frac{3}{4}$ inches stroke, designed to develop 24 horsepower at 800 revolutions, at which speed the propeller normally turns. The inlet valves are mechanically operated, and are situated in the cylinder head; they are worked by rocking levers from the same cam shaft that operates the exhaust valves. On the other side of the engine there is a second cam shaft, which operates the mechanical make-and-break of the low-tension magneto ignition. High-tension ignition is also provided as a standby; this is done to eliminate the chance of a motor stoppage due to ignition trouble in a seaway. A special feature of the engine is the very large bearing surface given to the main bearings and crank pins, the mean pressures on these being kept as low as standard marine practice, and consequently far lower than is usually the case in petrol engines. The engine is also extremely accessible. It is held to its bedplate by three lugs on each side in the form of hinge-joints, thus enabling it to be swung to one side or the other to get at the big ends or main bearings. Two large and instantly-detachable inspection doors are fitted to the base chamber, through which the hand may be put on to any big end or main bearing. The two cam shafts are held up to their bearings by caps, and these bearings can be removed from the rest of the engine without disturbing the cylinders or base chamber.

The crank shaft is also held up into the engine bed, so that when the engine is swung on one side, and the oil-bath casing removed, the big ends and main bearings are immediately accessible for adjustment.

By means of the design of the carburetter, which gives a positive mixture for every position of throttle opening, the engine will run and pull steadily at all speeds from 140 revolutions upwards. The torque is practically constant throughout the range.

The circulation of water is maintained by a centrifugal pump, which is carried low down forward and driven direct from the crank shaft by a roller chain. The water is deliv-

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ered first to a water-cooled exhaust pot carried close to the exhaust outlet, and from there proceeds to the engine jackets, and afterwards to the outlet overboard.

The valves are of very large diameter, of 25 per cent. nickel-steel. The crank shaft is a chrome-vanadium steel forging. From the engine the power is taken through a leather-faced cone clutch, which is engaged by an adjustable coil spring, and disengaged by a pedal. The clutch transmits the power to a small gear box, giving a forward and reverse gear by means of three miter bevels and a sliding sleeve with dogs, which enables the two shafts to be dogged together, or the power transmitted through the miter wheels. The power is transmitted through the bevels only when running astern. This gear box forms an oil-tight case, in which the bevels run, and which also lubricates the large phosphor-bronze bearings. It also has a platform cast on it, to which is fixed a cast-iron steering and control column. On this is mounted a vertical marine steering wheel, and in a convenient position for the steersman's right hand are two small handles, controlling, respectively, the throttle of the carburetter and the forward and reverse mechanism. The starting gear is also carried in this column. A removable handle fits into a socket in the column at a convenient height, and by means of a heavy chain and disengaging dogs the motor can be started. When once going the dogs disengage, and the chain does not run.

The after end of the engine housing forms an instrument board, on which is placed a pump lubricator, which is driven from a worm on the commutator spindle, and, consequently, delivers lubricant in quantities proportional to the speed of the engine. The two switches for magneto and high-tension ignition, the coil for the latter and the advance for both ignitions are also placed here.

Once the engine is started, the ignition may be advanced, and left without further adjustment during a run. Thus the whole control of the boat is easily managed by one man at the steering column, who can maneuver with great facility without moving from his place, and can re-start the motor, should he stop it. The engine is placed on its bearers with the

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crank-center line horizontal, so that the oil in the bottom half of crank case may be equally distributed to the forward and after pairs of cylinders. This necessitates a Hooke's joint behind the gear box to connect with the tail shaft. The joint is made with the center line of both pairs of pins in the same plane; the pins are of nickel-steel, running in phosphor-bronze bushes. This joint runs quite cool and gives no trouble. The tail shaft is of delta metal, and is fitted with two ball-bearing thrust blocks and a gun-metal three-bladed propeller 17 inches in diameter. The whole installation of engine, gear box, steering column and control is lined up and fixed to a bedplate consisting of 4-inch channels, thus making a complete unit in itself, which can be fitted in a boat without the more difficult and often unsatisfactory operation of lining up in place.

The boat was completed in July, and ran in the Motor Yacht Club's reliability trials. These trials consisted of two days' running of ten hours each without stop, and an allowance of only ten minutes between the two runs to attend to the motor. The course was mainly in Southampton water, but there was an optional open-sea course in the Solent, round which this boat ran.

The first day the *Iris* ran absolutely without any stop and with great regularity for the ten hours in a smooth sea.

The second day, owing to the driver omitting to keep his front tank filled, the motor stopped, but it took only four minutes to find the cause, pump in petrol, and re-start the motor, though it was very rough at the time. On this day it was so rough that the trials were stopped after five hours' running, and the boat showed her great seaworthiness on the open course. She was the only one of her class (30-footer) which could be driven through the seas at her full speed without slowing down at all, and, thus driven, shipped no green water, but only spray. The reduction in speed due to the rough water was only 6.3 per cent. over the calm day, whereas reductions of speed up to 43 per cent. were shown by some of the other boats.

Since the trials the boat has been repeatedly navigated

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single handed, and has proved herself exceedingly handy—so much so that, with a crew of only two, a tow-rope was passed to another boat which was towed across the Solent in rough water. Her speed is 10½ knots.

Fathomer.—The Coast Survey steamer *Fathomer* was recently placed in commission at Manila, and will be employed in the extensive surveying operations that are necessary to provide proper charts of the Philippine Islands.

The *Fathomer* was built by the Hongkong & Whampao Dock Co., at Hongkong, under a contract with the Philippine Government. The vessel was constructed under Lloyd's inspection as well as the inspection of the representative of the survey. Because of Lloyd's requirements nearly all the material had to be brought out from England. The contract was signed January 27, 1904; the trial trip was had on December 12, 1904, and the vessel was delivered in Manila Bay on January 19, 1905. The contract price of the vessel at Hongkong was $66,550, and, including delivery in Manila Bay, and extras but not equipment supplies, the total cost was about $70,000.

The *Fathomer* is a steel vessel of 368 gross tons; length over all, 160 feet; breadth, 25 feet; mean draught, 10 feet. The hull is divided into seven watertight compartments. In place of a double bottom, the lower decks, both forward and aft of the machinery space, are of steel and watertight, and there is special strengthening above Lloyd's requirements under the boiler and engine-room space. The mechanical equipment was made simple, because of the intention to employ mainly a native crew. A Scotch boiler and compound engine furnish the required speed of 10 knots. There is provided a dynamo, evaporator, small refrigerating machine and steam windlass.

Special attention was given to the boat equipment, an important feature in the success of surveying work. The *Fathomer* carries a 28-foot steam launch, a 24-foot whaleboat, a 20-foot cutter, a 16-foot dinghy and a 12-foot punt or flatboat. Wellin davits are provided for the launch, and the falls lead to the steam windlass.

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MERCHANT SHIPS.

Maheno.—The turbine-driven steamer *Maheno* has just been completed by Messrs. William Denny and Brothers, Dumbarton, for the Union Steamship Company of New Zealand, and is the second turbine-driven steamer possessed by them. Her dimensions are: Length, 400 feet; breadth, 50 feet; depth, 33 feet 6 inches molded to upper deck; and she has almost a complete shade deck, with a boat deck over a large portion thereof. She is elaborately fitted out for passengers, of whom she will carry 223 first class, 116 second and 60 third class. The trial had to be carried out with not less than 300 tons of dead weight on board, but there is capacity for more cargo. At her first official trial, at full power, with all boilers in use, the *Maheno* easily attained a mean speed of 17.5 knots, and at the second trial, which had to be done with a third of the boilers shut down, the speed maintained for six hours was 16.4 knots—considerably over the guarantee. The vessel is propelled by a set of Parsons patent turbines, constructed by Messrs. Denny and Co., Dumbarton. There are three turbines, one high and two low pressure, working three shafts, with three propellers in all. The go-astern turbines are contained within the low pressure, and work on the wing shafts. The reversing gear is particularly well arranged, the whole of the engines being easily controlled by one engineer. Steam is supplied by four cylindrical tubular boilers, two double-ended and two single, at a working pressure of 150 pounds per square inch, and fitted with Howden's system of forced draft. The auxiliary machinery is very complete, and consists of air and circulating pumps to each main condenser, each set consisting of two Edwards air pumps and one centrifugal pump, driven by simple two-cylinder engines; two of Weir's feed pumps, auxiliary condenser with necessary pumps, two large duplex pumps for bilge and ballast purposes, a Caird and Raynor distiller, and a vertical duplex wash deck and

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fire engine, suitable for working See's ash ejector, sanitary pump, steward's pump, water-service pump, oil pumps, &c.—"Engineering," London.

Princesse Elisabeth.—A new turbine steamer has been added to the cross-Channel fleet, under the Belgian flag, for running between Dover and Ostend. This vessel, the *Princesse Elisabeth*, 350 feet long by 40 feet beam, was constructed at Antwerp, and successfully went through her trials on the Clyde, between the Cloch and the Cumbrae Lights, and maintained an average speed thereon of twenty-four knots on four runs. The vessel marks another step in advance made in turbine propulsion, and the speed obtained holds the record for this class of steamer. The hull of the vessel, boilers and parts of the machinery have been built by the John Cockerill Company; the turbine machinery was constructed at Wallsend, under contract with the Parsons Foreign Patents Company, Limited; the auxiliary machinery, wet and dry-air pumps, circulating, feed, oil, sanitary and bilge pumps, have all been supplied by Messrs. G. and J. Weir, Limited, of Glasgow.

Ferryboat Brooklyn—*Trial.*—The *Brooklyn* was the second to be tried of the four ferryboats built by the Maryland Steel Company for the City of New York.

The four ferryboats are the same size and built from the same plans. They are of the double-end type, having propeller at each end.

Length over all, 250 feet; length between perpendiculars, 246 feet; beam over guards, 66 feet; beam molded, 46 feet; depth, 19 feet 6 inches; draught, 12 feet; displacement at 11 feet 10 inches; draught, 1,394 tons.

Engines.—There are two compound engines, set with the L.P. cylinders adjacent, the crank shafts being coupled between the L.P. cylinders, and the shafting continues from end to end of the vessel.

The engines were designed for a working pressure of 175 pounds at the H.P. valve chest. Designed piston speed, 750 feet. The engines are same size, $\frac{22\frac{1}{2} \text{ ins.} \times 50 \text{ ins.}}{30}$.

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The back frames of engines are cast and of inverted Y type. The front columns are forged. The engines are simple, easily accessible and substantial in their design and construction.

Boilers.—There are four Babcock & Wilcox boilers, located two in compartment forward and two in compartment abaft engine compartment.

Total G. S., square feet	340
H. S., square feet	12,500
Working pressure, pounds	225

The pressure is reduced to 175 pounds at the engine.

Condenser.—There is one cylindrical built-up condenser with shell of $\frac{3}{8}$-inch steel plate.

Diameter of tubes, inch	$\frac{3}{4}$
Thickness of tubes, B.W.G.	18
Total cooling surface, square feet	4,500

The cooling water makes a double pass, entering the top and returning through the bottom tubes.

This arrangement gives warmer feed water and requires less circulating water than where the cooling water passes through the bottom tubes first and leaves the condenser from top nest of tubes.

Pumps.—There is one centrifugal circulating pump supplying the condensing water. It is fitted to take water from a side, a bottom and a bilge injection. The bottom injection is to be used when the floating ice prevents the use of the side injection.

There is one twin air pump of Cameron make $\frac{10'' \times 22''}{18''}$.

There are two main feed pumps, one in each fireroom, and an auxiliary feed pump in the engine room. These pumps are of the Cameron vertical simplex type, the main feed pumps being $\frac{10'' \times 6''}{12''}$ and the auxiliary $\frac{12'' \times 6''}{16''}$. Besides the feed pumps there is a $2\frac{1}{2}$-inch Metropolitan injector in each fireroom for feeding the boilers.

The auxiliary feed pump alone was in use to supply water

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to all boilers during the trial. It did this work easily and quietly, making 26 double strokes per minute.

There is one Cameron vertical simplex sanitary pump $\frac{6'' \times 8''}{8''}$, and one Cameron duplex fresh-water pump $\frac{4\frac{1}{2}'' \times 3\frac{3}{4}''}{4''}$.

There is a $\frac{12'' \times 9\frac{3}{4}''}{10''}$ Blake fire and bilge pump in the engine room.

Propellers.—There are two right-hand, four-bladed propellers.

Diameter, feet	11
Pitch, feet and inches	14- 4
Helicoidal area, square feet	47.66
Projected area, square feet	39.55
Ratio of projected area to helicoidal area	1 : 1.23
helicoidal area to disc area	.501

The trial was made in the Chesapeake Bay. The averages are as follows:

Steam pressure at engines, pounds	176
Recorded pressure (gauge), pounds	29
Vacuum, inches	24.3
Revolutions per minute, main engines	147.5
air pump	46.4
circulating pump	275
I.H.P., both engines	4,093
Speed in knots	15.93
Slip of propeller, per cent	24.3
Draught on trial, feet and inches	11-10
Displacement, tons	1,394
Wetted surface, square feet	9,066
I.H.P. per 100 square feet of wetted surface at speed of 10 knots	11.25
Admiralty coefficient	134.2

The boilers were operated under one-inch air pressure.

No account was taken of the weight of coal, so that the economical performance is not known. The operation, however, of the engines, boilers and all auxiliaries, as observed on the trial, was satisfactory.

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OBITUARY.

LIEUTENANT COMMANDER ROBERT I. REID, U. S. N. (RETIRED).

It is with extreme regret that the Council announces the death of one of its members since the last issue of the JOURNAL.

Lieutenant Commander Robert I. Reid, U. S. N. (Retired), died at Caldwell, N. J., November 10, 1905.

BOOKS RECEIVED.

DAVID G. FARRAGUT.—By JOHN RANDOLPH SPEARS, published by GEORGE W. JACOBS & COMPANY, Philadelphia, Pa.

This is the third of the American Crisis Biographies, and gives in a comprehensive and interesting manner a description of those actions, both before and during our Civil War, in which Admiral Farragut took such a conspicuous part.

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NOTES OF THE SOCIETY.

A regular meeting of the Society was held at the Navy Department, October 3d, at which nomimations were made for Officers of the Society for the coming year, as follows:

For President.

Commander A. B. CANAGA, U. S. N.
Commander W. F. WORTHINGTON, U. S. N.
Lieutenant Commander B. C. BRYAN, U. S. N.

For Secretary-Treasurer.

Lieutenant Commander T. C. FENTON, U. S. N.
Lieutenant C. K. MALLORY, U. S. N., Retired.

For Members of Council.
(Three to be selected.)

Rear Admiral G. W. BAIRD, U. S. N., Retired.
Engineer-in-Chief C. A. MCALLISTER, U. S. R. C. S.
Commander W. M. PARKS, U. S. N.
Commander R. S. GRIFFIN, U. S. N.
Lieutenant Commander B. C. BRYAN, U. S. N.
Commander W. W. WHITE, U. S. N., Retired.
Lieutenant A. M. COOK, U. S. N.
Civil Engineer A. C. CUNNINGHAM, U. S. N.
Lieutenant JOHN HALLIGAN, JR., U. S. N.
Lieutenant H. T. WINSTON, U. S. N.

Voting slips have been forwarded to all the members, to be returned at once. The votes will be counted at the Annual Meeting of the Society, to be held in the Bureau of Steam Engineering, Navy Department, at 4.30 P. M., Thursday, December 28, 1905.

At a meeting of the Council, held in the Bureau of Steam Engineering, October 27, 1905, Lieutenant C. K. Mallory, U. S. N., tendered his resignation as Secretary-Treasurer, and Lieutenant Commander Theo. C. Fenton, U. S. N., was elected to succeed him.

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OFFICERS OF THE SOCIETY.

President:

Captain A. F. Dixon, U. S. Navy.

Secretary-Treasurer:

Lieutenant Commander Theo. C. Fenton, U. S. Navy.

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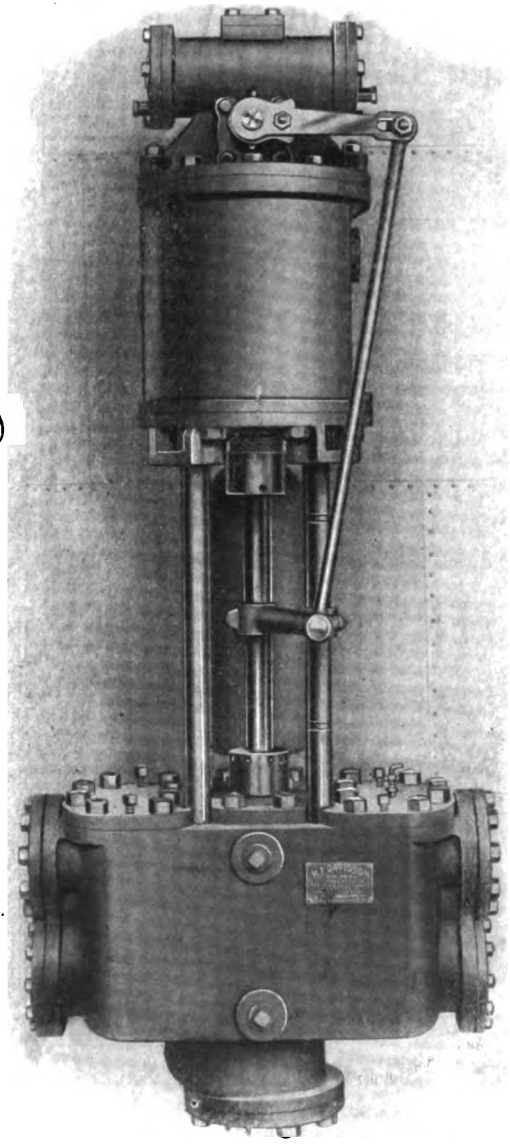

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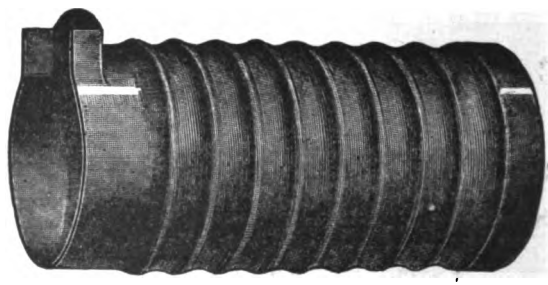

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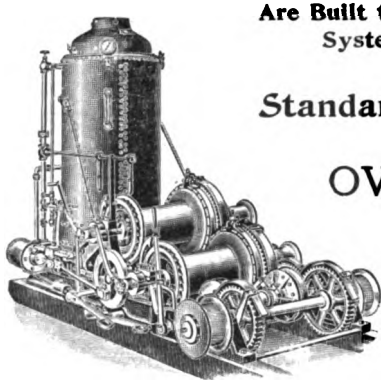

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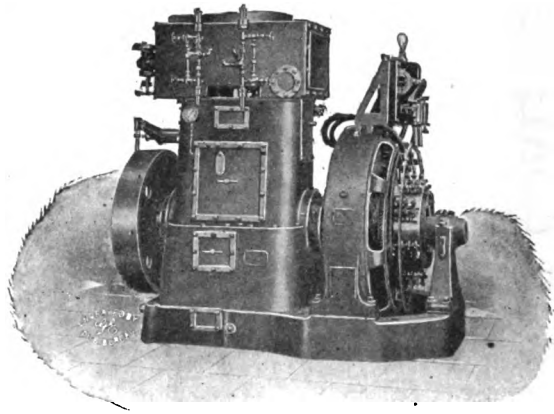

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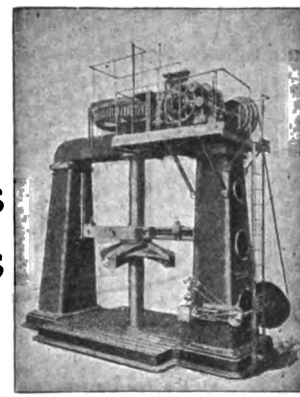

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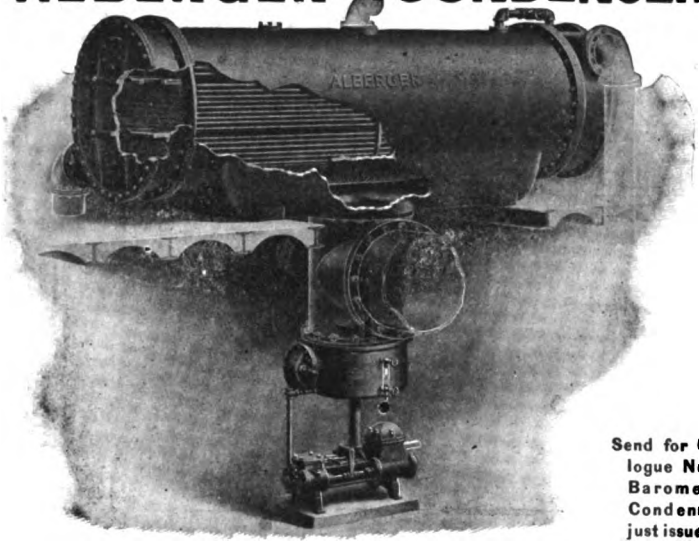

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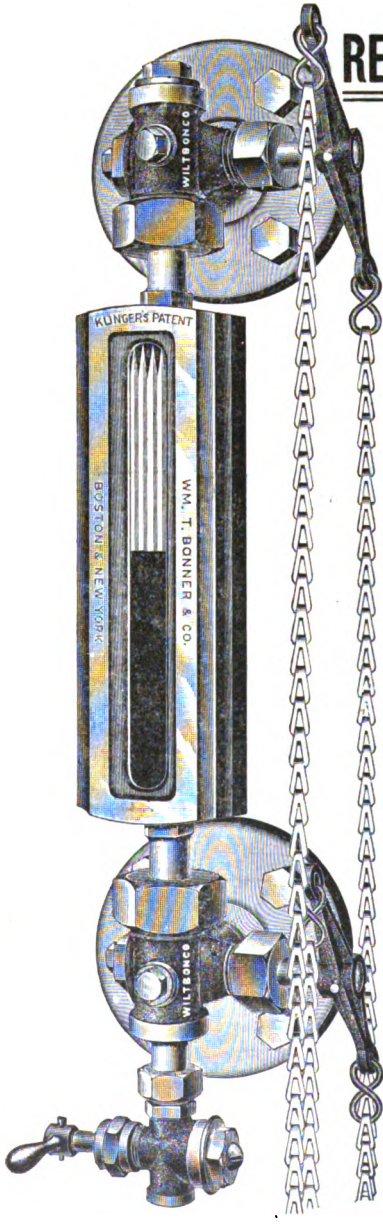

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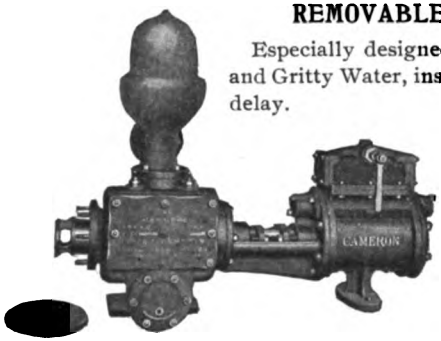

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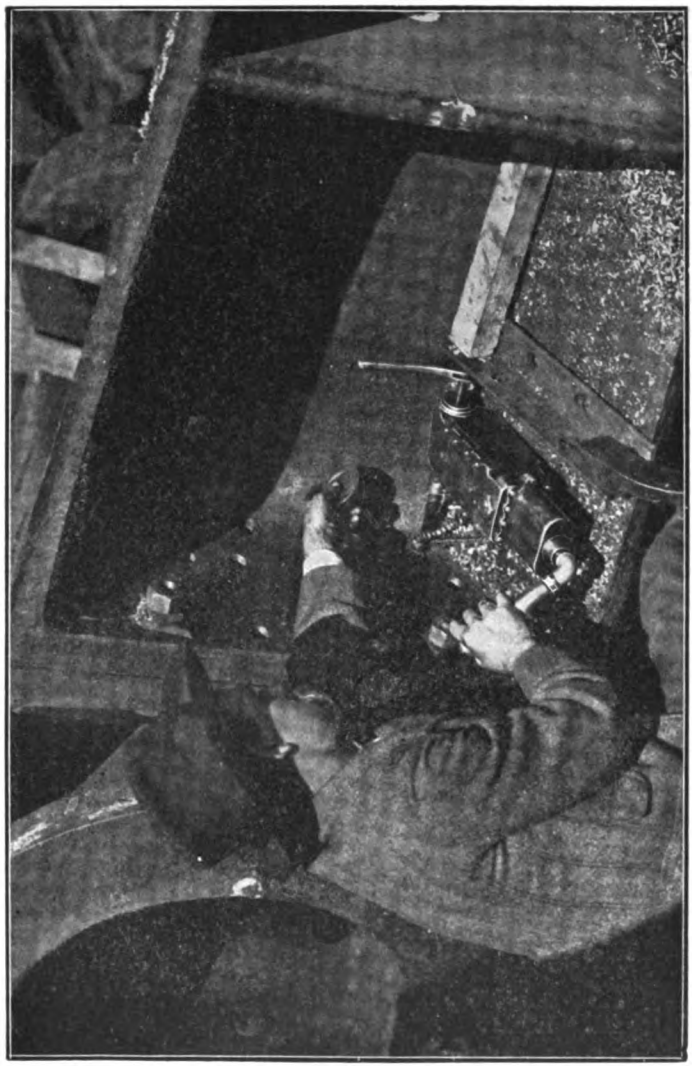

No. 2 U. & W. Piston Air Drill at The Cramp Ship and Engine Building Co. at work on stern post of the U. S. S. "Tennessee."

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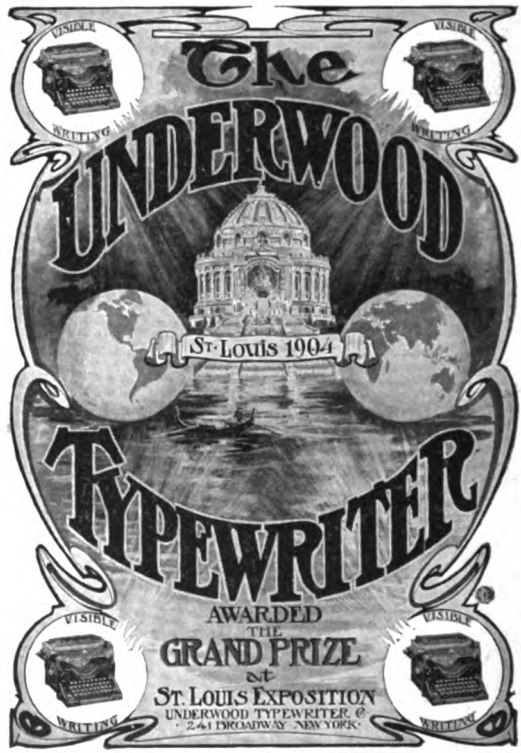

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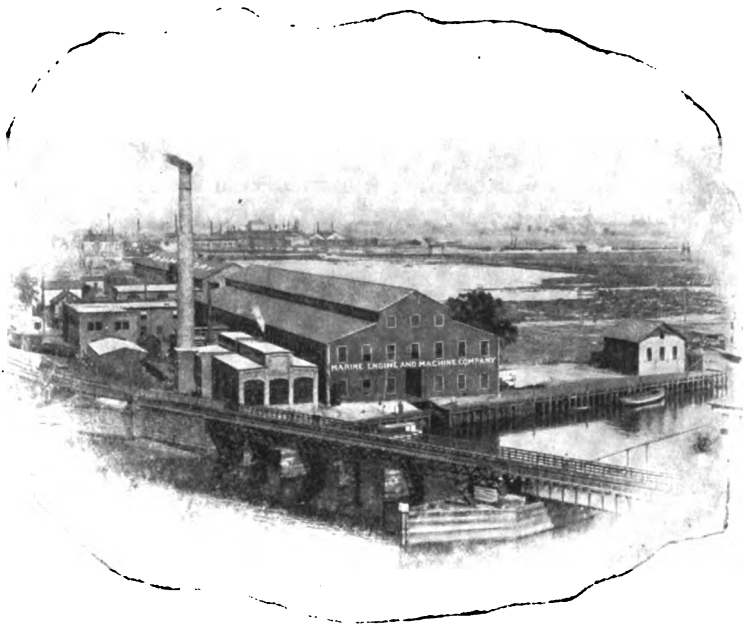

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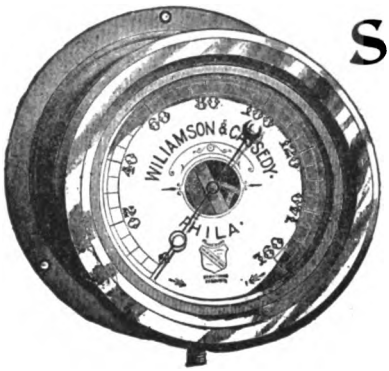

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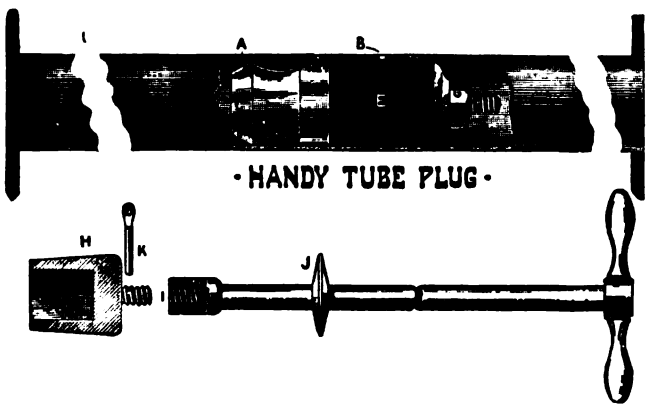

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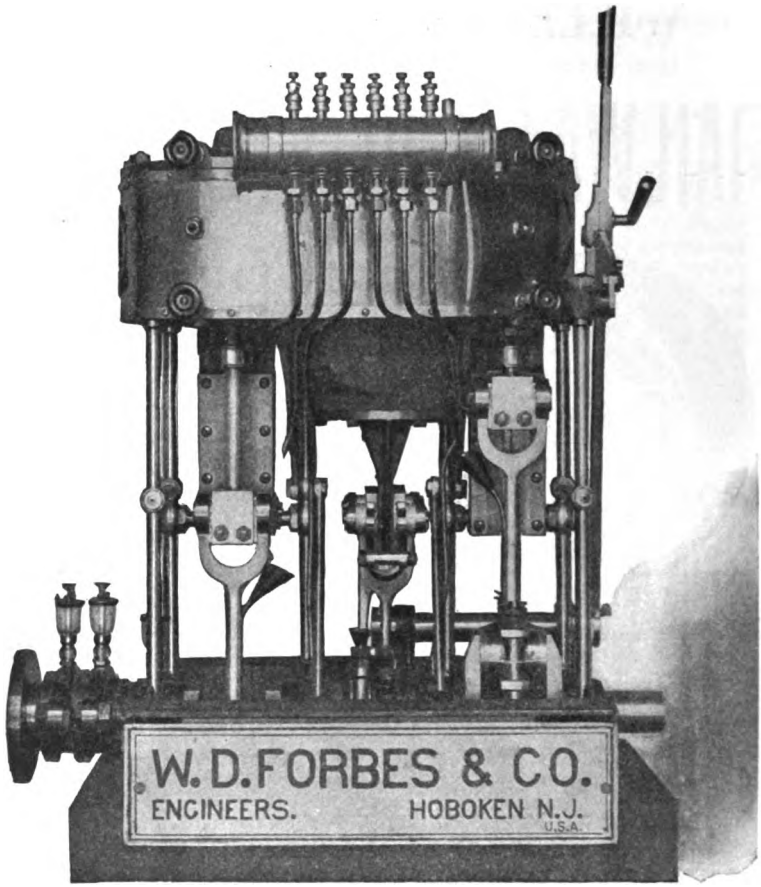

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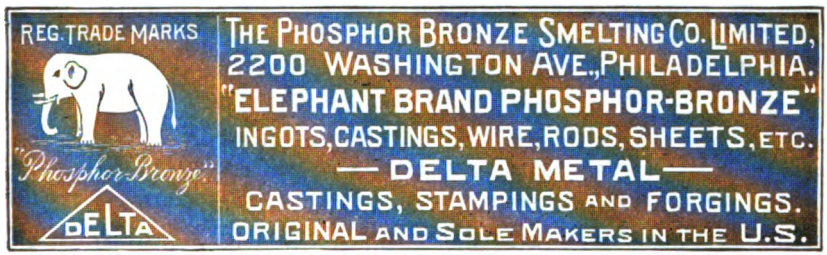

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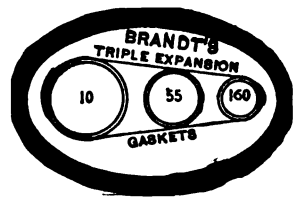

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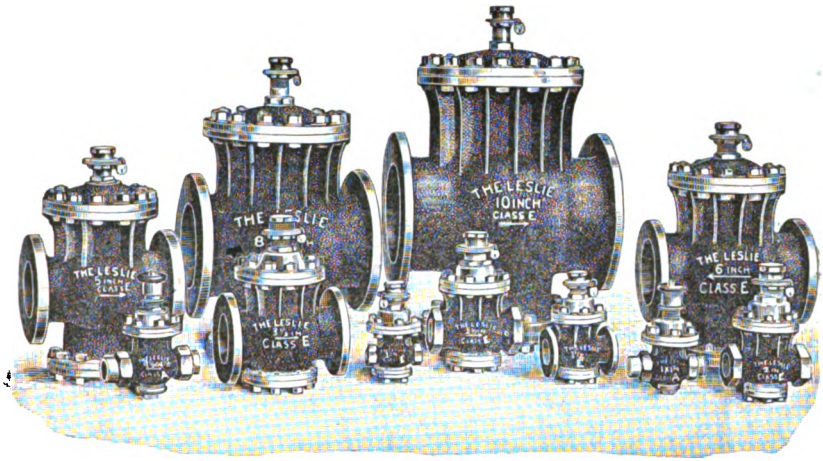

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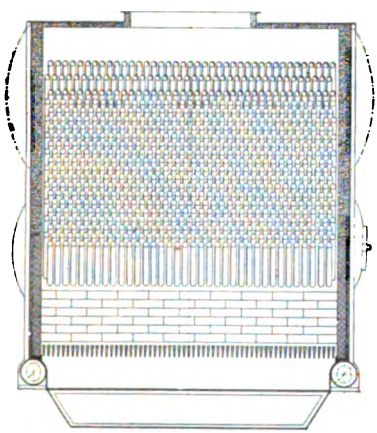

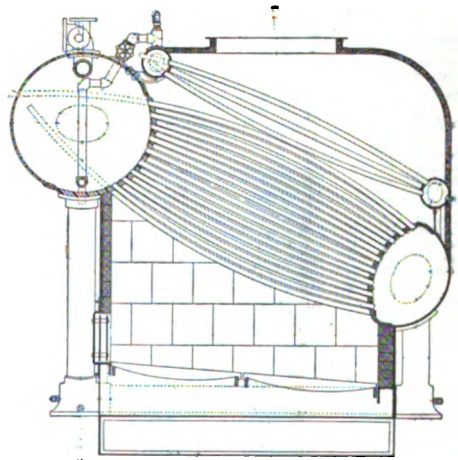

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FOR HEATING AND CIRCULATING THE WATER IN STEAM BOILERS. NEARLY 500 INSTALLED.

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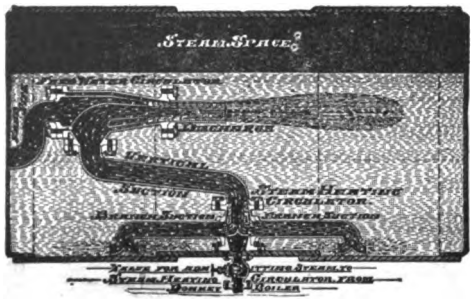

Increases evaporation 5 to 15 per cent. Prevents Corrosion.
Will pay for itself in a year with saving in repairs.
Keeps all parts of boiler at an even temperature, and prevents leaks in bottom seams.
Has no extra joints to leak.
Creates a constant automatic circulation as long as boiler is fed.

The Steam-Heating Attachment will heat and circulate the water with steam from donkey boiler in a half hour, while fires are being started, and be ready for steam with less straining than slow fires burning twelve hours.

270 to 320 degrees temperature at bottom of boiler.

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FOR INDUCED DRAFT. NEARLY 400 JETS.

ALL SIZES, FROM 7 INCHES TO 8 FEET, HAVE BEEN BUILT.

Powerful. Economical. 15 to 25 Per cent. Increase in Power.
Standard Jet Gives ¾ to 1⅛-Inch Draft in Stack.
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PRICES MODERATE. REQUIRES NO ATTENTION.

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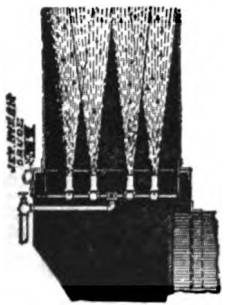
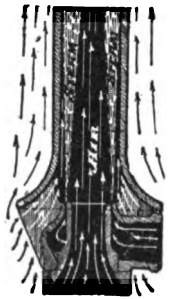


SECTION OF STACK.



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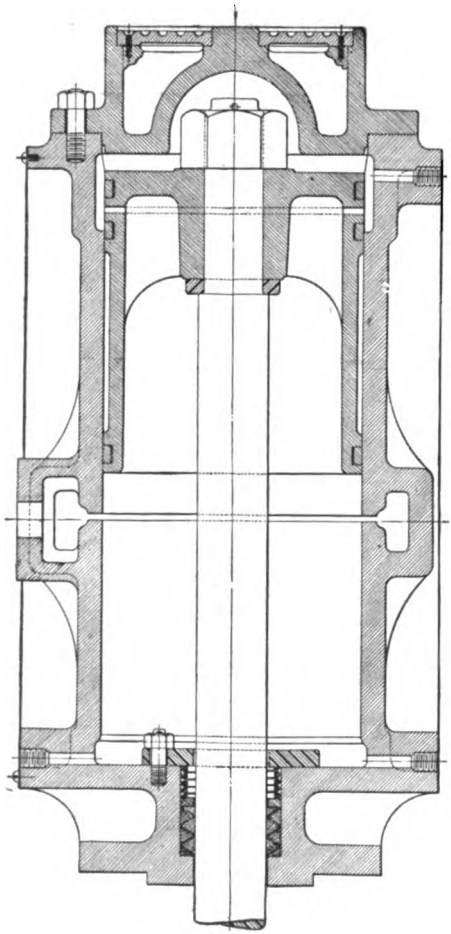
LOVEKIN IMPROVED ASSISTANT CYLINDER FOR VALVE GEARS (PATENTED).



The Lovekin-Thom Improved Assistant Cylinder as Fitted on S. S. "Manchuria."

See papers read before Society of Naval Architects and Marine Engineers, Nov., 1902. Prize awarded for this article. Also see JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. XVI, No. 3, and Vol. XVII, No. 3.

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It floats the entire valve gear.

HOW?

By balancing the weight of the valve gear, and the forces due to inertia.

WHAT IS THE RESULT?

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3d.—Smoother running of engines.

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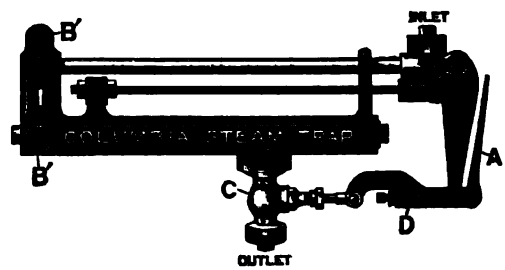
Always open when water is at the discharge valve.

Always closed when steam is at the discharge valve.



No Floats or Inside Levers. Outlet valve same area as the pipe. No other kind of Steam Trap has this.

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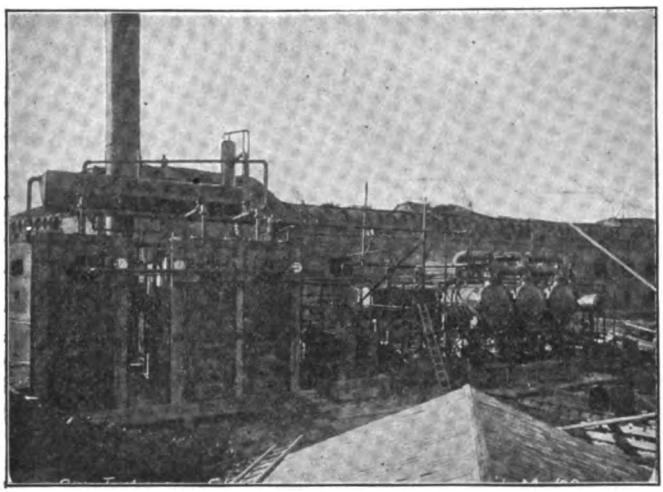
FOR THE ECONOMICAL MANUFACTURE OF

DISTILLED WATER FOR ALL PURPOSES.



Plant for Making Distilled Water from Sea Water, Dry Tortugas, Fla.

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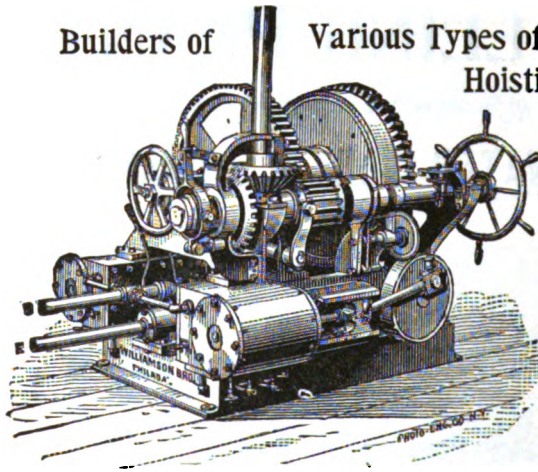
Builders of Various Types of Steam Steering, Hoisting and Winding Engines, Evaporators, Distillers, Steam Cranes, Etc.



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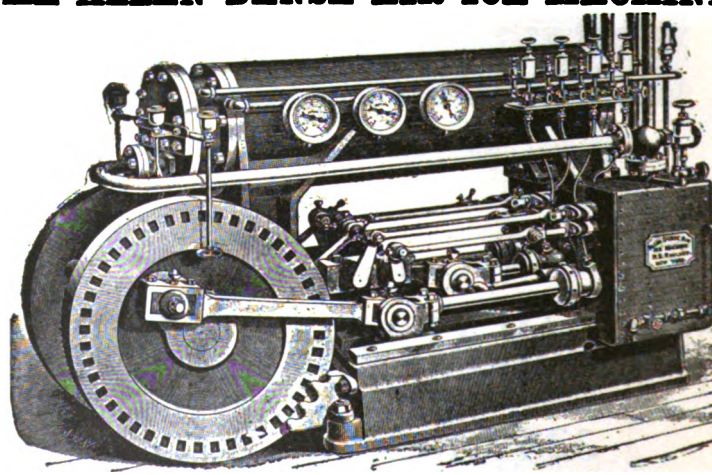

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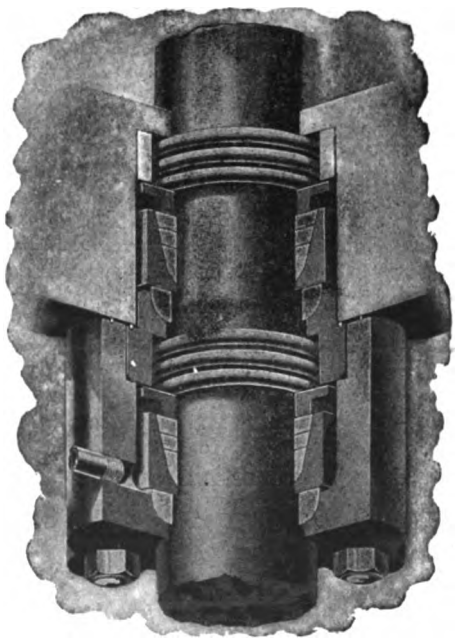

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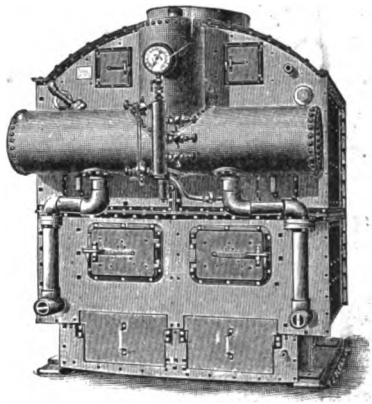

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OVER 250 STEAM VESSELS NOW EQUIPPED.

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WHEN LIGHT WEIGHT, SMALL SPACE, DURABILITY, ECONOMY, ACCESSIBILITY, HIGH PRESSURE, SAFETY, ARE CONSIDERED, IT OUTCLASSES ALL OTHERS.

OPERATES NICELY IN BATTERIES.
AUTOMATIC FEED REGULATION.

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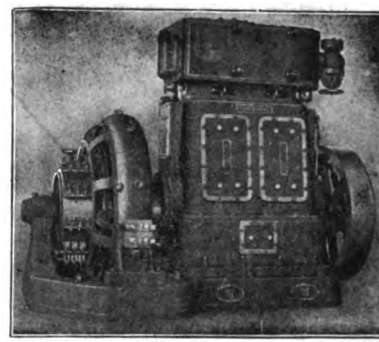

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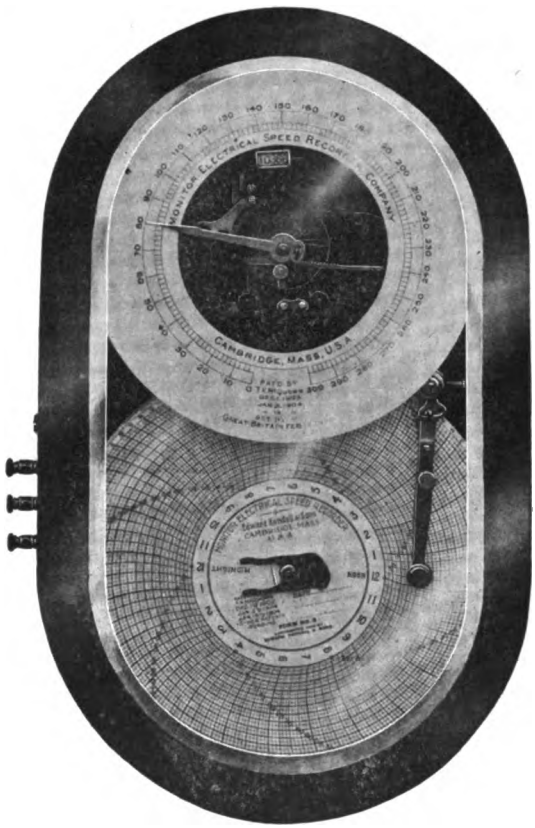

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AND PARTICULARLY SPECIAL DESIGNS AND THEIR CONTAINED INSTALLATION ADAPTING THEM FOR

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No. 557,835, April 7th, 1896.
No. 575,890, January 26th, 1897.
No. 581,213, April 20th, 1897.
No. 591,851, October 19th, 1897.
No. 611,636, October 4th, 1898.
No. 617,750, January 17th, 1899.
No. 638,342, December 5th, 1899.
No. 650,758, May 29th, 1900.
No. 652,970, July 3d, 1900.
No. 676,820, June 18th, 1901.
No. 695,215, March 11th, 1902.
No. 709,335, September 16th, 1902.
No. 710,472, October 7th, 1902.
No. 712,814, November, 4th, 1902.
No. 714,921, December 2d, 1902.
No. 715,395, December 9th, 1902.
No. 716,059, December 16th, 1902.
No. 716,844, December 23d, 1902.
No. 717,101, December 30th, 1902.
No. 719,235, January 27th, 1903.
No. 725,570, April 14th, 1903.
No. 726,227, April 21st, 1903.
No. 726,705, April 28th, 1903.
No. 726,947, May 5th, 1903.
No. 738,725, September 8th, 1903.
No. 754,222, March 8th, 1904.
No. 759,622, May 10th, 1904.
Other patents pending.

These patents cover broadly the principal features embodied in the **LAKE TYPE** of **submarine boat,** the methods of construction, of installing the machinery and auxiliary apparatus, etc.

Among other features, they cover :

The method of utilizing the bottom of the sea as a guiding medium ; the diving compartment and means to enable divers to leave and enter the vessel while submerged ; the superstructure and means for storing fuel and supplies within the same ; the invisible conning tower ; means to prevent injurious gases from leaking into the boat ; means to enable operators in the boat to secure an all around view of the horizon while submerged ; means for navigation beneath fields of ice ; means to automatically regulate the depth of submergence by the use of hydroplanes and otherwise ; means for increasing stability and maintaining trim ; special methods of installing storage batteries ; apparatus for handling and discharging torpedoes under water ; means of control of propelling apparatus ; method of establishing communication with the shore while at sea ; anchor operating mechanism to enable the vessel to remain for long periods at any desired depth, etc., etc.

These patented devices are the result of 20 years of study and experiment with submarine boats and are covered by over 300 patent claims.

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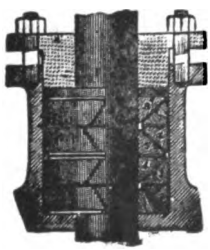

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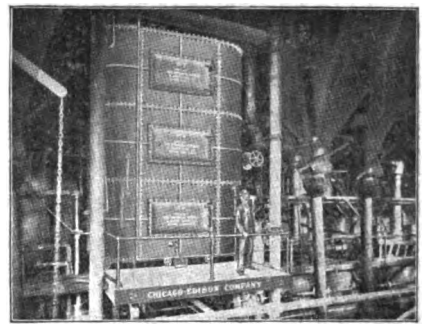

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